MACHINERY.

October, 1905.

FANS.—1.

CHARLES L. HUBBARD.

There are two types of fans in common use known as the centrifugal fan or blower, and the disk fan or propeller.

The former consists of a number of straight or slightly curved blades extending radially from an axis as shown in Fig. 1. When the fan is in motion the air in contact with the blades is thrown outward by the action of centrifugal force and delivered at the outer circumference or periphery of the wheel. A partial vacuum is thus produced at the center of the wheel, and air from the outside flows in to take the place of that which has been discharged. Fig. 2 illustrates the action of a centrifugal fan, the arrows showing the path of the air.

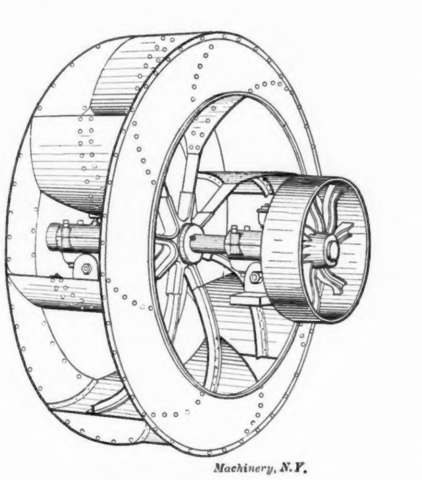


Fig. 1. Centrifugal Fan.

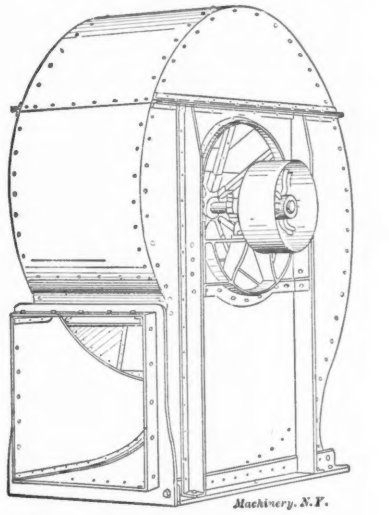


Fig. 3. Centrifugal Fan and Casing.

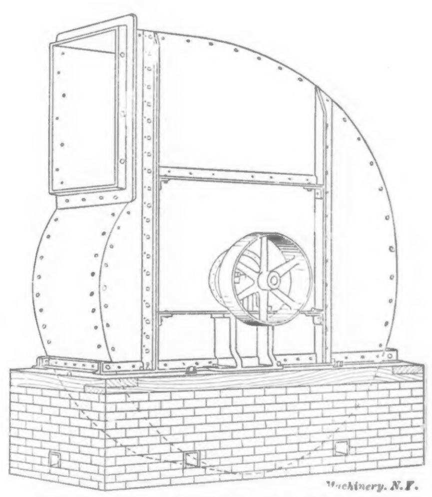


Fig. 4. Fan partly Encased by Brick Wall

This type of fan is usually enclosed in a steel plate casing of such form as to provide for the free movement of the air as it escapes from the periphery of the wheel.

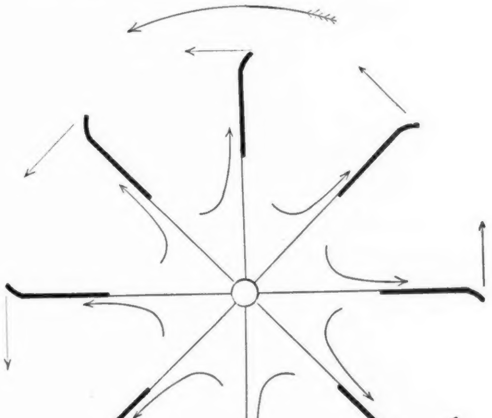


Fig. 2. Direction of Flow in Centrifugal Fan.

An opening in the circumference of the casing serves as an outlet into the distributing ducts which carry the air to the various rooms to be ventlated, or to the furnaces in the case of mechanical draft.

A fan with casing is shown in Fig. 3. The discharge opening can be placed in any position desired, either up, down, top horizontal, bottom horizontal, or at any angle. Where the height of the fan room is limited, a form called the three-quarter housing may be used, in which the lower part of the casing is replaced by a brick pit below the floor level (see Fig. 4). Another form of the centrifugal fan is shown in Fig. 5. This is known as the cone fan and is commonly placed in an opening in a brick wall and discharges air from its entire periphery into a room called a plenum chamber with which the various distributing ducts connect. This fan is often made double by placing two wheels back to back and surrounding them with a steel casing in a similar manner to the one shown in Fig. 3.

Cone fans are very efficient and are capable of moving large quantities of air at moderate speeds. Fig. 6 shows a form of small direct-connected exhauster commonly used for ventilating toilet rooms, chemical hoods, etc., and for furnishing a forced draft for forges and small boilers. Centrifugal fans are used almost exclusively for supplying air for the ventilation of buildings, for forced blast heating and for mechanical draft. They are also used as exhausters for removing the air from buildings when the resistance is considerable, and the quantity of air to be handled is large.

The disk fan is similar in construction to the propeller of a vessel and moves the air in lines parallel to its axis. This fan is made in various forms with both flat and curved blades. Fig. 7 shows one of the various designs arranged either for belted or direct-connected motor. This type of fan is light in construction, requires but little power at low speeds and is easily erected. It is especially adapted to exhaust ventilation when the resistance is small, being conveniently placed in the attic or upper part of a building and driven by an electric motor. Disk fans are largely used for the ventilation of public toilet rooms, smoking rooms, restaurants, etc., and are often connected with the main vent flues of large buildings, such as schools, halls, churches, theatres, etc. They are especially adapted for use in connection with gravity heating systems where the flow of air through vent flues is apt to be sluggish in mild weather.

Theory of Centrifugal Fans.

The action of a fan is affected to such an extent by the various conditions under which it operates that it is impossible to give fixed rules for determining the exact results to be expected in any particular instance. This being the case, it seems best to take up the subject briefly from a theoretical standpoint and then show what corrections are necessary in the case of a given fan under actual working conditions. As al-

ready stated, the rotation of a fan of this type sets in motion the air between the blades, which by the action of centrifugal force is delivered at the periphery of the wheel into the casing surrounding it. As the velocity of flow through the discharge outlet depends upon the pressure or head within the casing, and this in turn upon the velocity of the blades, it becomes necessary to examine briefly into the relations existing between these quantities.

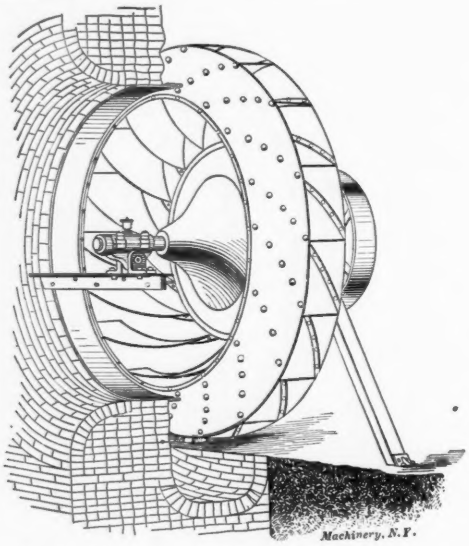


Fig. 5. Cone Fan Placed in Opening in Brick Wall.

If a vessel as shown in Fig. 8 be filled with water, a certain pressure will be exerted upon the bottom, depending upon the depth and temperature of the water. If the weight of a cubic inch of water at a temperature of 50 degrees is .036 of a pound (called its *density* at that temperature) and the depth of the water in the vessel is 20 inches, then the pressure upon each square inch of the bottom will be the weight of a column of water having a sectional area of one square inch and a height of 20 inches, which in the above case is $20 \times .036 = .72$ of a pound; so that for general use we may write $p = h \times d$, or $h = \frac{p}{d}$, in which

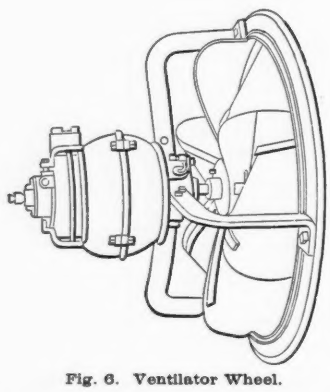


Fig. 6. Ventilator Wheel.

h = the height of the column of water, called the head.
d = the density of the water.
p = the pressure produced.

If h is taken in inches and d in ounces per cubic inch, then p will be in ounces per square inch. If h is in feet and d in pounds per cubic foot, p will be in pounds per square foot, and so on, depending upon the units taken. When dealing with water pressure it is customary to take such units as will give p in pounds per square inch.

In the case of air pressure in connection with fans, ounces per square inch is commonly used.

If a pipe be inserted in the side of the vessel (Fig. 9) at any given distance from the surface of the water, and a supply of water be provided sufficient to keep the level constant, a pressure of $h \times d$ will be exerted at the entrance to the pipe, causing the water to flow through it. The height h in this case is called the total head producing flow, and is divided into three parts, as follows: The *entry head*, that required to overcome the resistance to entry into the pipe; the *friction* or *pressure head*, that required to overcome the resistance due to the friction of the water in the pipe, and the *velocity head*, which is that used in giving motion or velocity to the water flowing through the pipe.

The entry head depends upon the form of entrance to the pipe, and with smooth rounded edges is inappreciable. The friction head depends upon the length and size of the pipe, the interior surface, the number of bends and the quantity of water flowing through it. The velocity head is the same as the height through which a body must fall in a vacuum to acquire

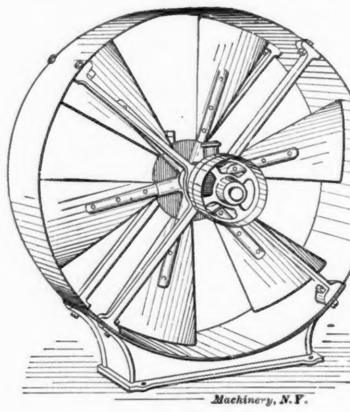


Fig. 7. Disk Fan.

the velocity with which the water flows into the pipe. This is given by the formula for falling bodies.

$$h = \frac{v^2}{2g} \text{ in which}$$

h = the head in feet,
v = the velocity in feet per second,
g = the acceleration due to gravity, and is equal to 32.16.

This may also be written in the form of $v = \sqrt{2\,g\,h}$.

In applying this to the flow of air from a fan casing, it is customary to consider only the last two, and if the outlet is short and properly formed, the friction head may be neglected also. When the fan is to be used for moving air through ducts and flues in its practical application to ventilation, the effect of frictional resistance is added and must be provided for as stated later. We have seen that $v = \sqrt{2\,g\,h}$, and that $h = \frac{p}{d}$. Substituting $\frac{p}{d}$ for h in the above formula, we have $v = \sqrt{2\,g\,\frac{p}{d}}$ in which v is the velocity in feet per second of a liquid flowing from one chamber into another, where the difference in pressure is p and the density is d.

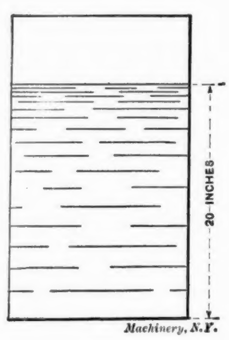


Fig. 8.

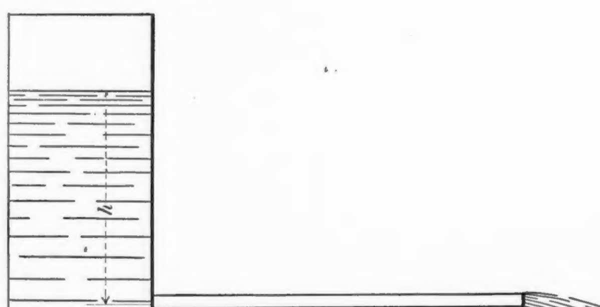


Fig. 9.

Applying this to the case of dry air at a temperature of 50 deg. F, and allowing for the change in density as it passes from a higher to a lower pressure the formula becomes for the small differences in pressure employed in ventilating work,

$$v = \sqrt{\frac{1746659 \times p}{235 + p}}$$

in which v is in feet per second, and p in ounces per square inch.

Table I. computed from the above formula is taken from "Mechanical Draft," published by the B. F. Sturtevant Co., and gives the velocity of dry air at a temperature of 50 deg. F. flowing into the atmosphere under different initial pressures.

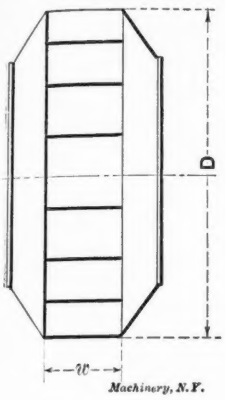


Fig. 10,

A simple approximate formula giving very near the same results for air at 50 deg. is $v = 65.5 \sqrt{h}$, in which v is the velocity in feet per second as before, h the pressure expressed in inches of water as indicated by the balanced height of a column of that liquid in a water gage. Pressure in inches of water column may be reduced to ounces per square inch by multiplying by .58.

For example: A pressure of ¼ of an ounce will produce a velocity of 43.08 feet per second, and a pressure of 1 ounce will produce a velocity of 86.03 feet per second, and so on.

The pressure within a fan casing is caused by the air being thrown from the tips of the blades, and varies with the velocity of rotation, that is, the higher the speed of the fan the greater will be the pressure produced.

When the various dimensions of a fan and casing are properly proportioned, the velocity of air-flow through the outlet will be the same as that of the tips of the blades, and the pressure within the casing will be that corresponding to this velocity. From this, it is evident that by knowing the diameter and speed of any given fan we can determine the peripheral velocity and find at once from Table I. the pressure produced within the casing.

TABLE I.

Pressure in Ounces per square inch.	Feet per Second.	Feet per Minute.
¼	43.08	2585.0
⅜	52.75	3165.1
½	60.90	3653.8
⅝	68.07	4084.0
¾	74.54	4472.6
⅞	80.50	4829.7
1	86.03	5161.7
1¼	96.13	5768.0

Velocity of dry air at 50° temperature.

Blast Area.

When the outlet from a fan casing is small, the air will pass out with a velocity equal to that of the tips of the blades, and the pressure within the casing will be that corresponding to the tip velocity. Now if the opening be slowly increased, while the speed of the fan remains constant, the air will continue to flow with the same velocity until a certain size is reached. The pressure in the casing will now begin to drop and the velocity of outflow become less than the tip or peripheral velocity. The effective area of outlet at the point when this change begins to take place is called the *capacity area* or *blast area* of the fan. This varies somewhat with different types and makes of fans, but for the common form of blower it is approximately one-third of the projected area of the fan opening at the periphery. That is $\frac{D\,w}{3}$, in which D is the diameter of the fan wheel and w its width at the circumference (see Fig. 10).

Table II. gives the speed of fans of different diameters necessary to maintain various pressures over an effective area equal to, or less than the blast area of the fan. The speeds given are in revolutions per minute, and are taken from "Mechanical Draft."

As a matter of fact the outlet of a fan casing is always made larger than the blast area, so that in actual practice the figures given in the following table must be corrected by certain factors as explained later.

Theoretical Capacity.

If we assume the effective outlet area of a fan to be equal to the blast area, its capacity at any given speed can be computed as shown in the following example. A fan 5 feet in diameter has a width of 2 feet at the tips of the blades. What quantity of air will it discharge at a speed of 200 revolutions per minute?

Taking the blast area as $\frac{D\,w}{3}$, we find it to be $\frac{5 \times 2}{3} = 3.33$ square feet.

At a speed of 200 revolutions the tip velocity of the fan will be $3.1416 \times 5 \times 200 = 3{,}141$ feet per minute.

TABLE II.

Diam. of Fan Wheel in feet.	Pressure in Ounces per Square Inch.								
	¼	⅜	½	⅝	¾	⅞	1	1¼	1½
2′	411	504	582	650	712	769	822	918	1005
2½′	329	403	465	520	570	615	657	734	804
3′	274	336	388	433	475	513	548	612	670
3½′	235	288	332	372	407	439	469	525	574
4′	206	252	291	325	356	384	411	459	502
4½′	183	224	258	289	316	342	365	408	447
5′	164	202	232	260	285	308	329	367	402
6′	137	168	194	217	238	256	274	306	335
7′	117	144	166	186	203	220	235	262	287
8′	103	126	146	163	178	192	205	230	251
9′	92	112	129	144	158	171	183	204	223
10′	82	101	116	130	142	154	164	184	201

R. P. M. required to produce given pressure with given wheel.

Therefore the air delivered by the fan is $3.33 \times 3{,}141 = 10{,}459$ cubic feet per minute.

What will be the capacity of the same fan working under a pressure of ½ ounce, and what will be the required speed? Looking in Table I. we find the velocity corresponding to ½ ounce pressure to be 3,653.8 feet per minute; therefore, the capacity of the fan is $3.33 \times 3{,}653.8 = 12{,}167$ cubic feet per minute.

The required speed may be taken directly from Table II., where it is found to be 232 revolutions per minute.

Power Required to Move Air.

The work done by a fan in moving air is represented by the pressure exerted multiplied by the distance through which it acts. This is expressed in foot pounds by the equation $W = PAV$, in which

W = work done, in foot-pounds per minute,

P = pressure at discharge opening, in pounds per square foot,

A = area in square feet, over which pressure P is exerted,

V = the velocity of flow through discharge outlet, in feet per minute.

TABLE III.

Pressure in Ounces per Square Inch.	Cubic feet of Dry Air at 50 degrees Temperature which will be Discharged through an Orifice having an Effective Area of 1 square inch.	Horse Power required to move the Given Volume of Air under the Given Conditions.
¼	17.95	.00122
⅜	21.98	.00225
½	25.37	.00346
⅝	28.36	.00483
¾	31.06	.00635
⅞	33.54	.00800
1	35.85	.00978
1¼	40.06	.01366

Volume of air discharged and horse power required to move same.

The horse power required for moving air through any given area of discharge is given by the formula

$$\text{H. P.} = \frac{d\,a\,v^3}{5\,100\,480} \text{ in which}$$

d = density of the air at the given temperature,

a = effective area of discharge outlet, in square inches,

v = velocity of flow, in feet per second.

Table III. (from "Mechanical Draft") gives the volume of air in cubic feet, which will be discharged per minute through an effective area of 1 square inch under different pressures; also the H. P. required for moving these quantities of air

under the different conditions. This table gives only the power necessary for *moving* the air, and does not take into consideration the *friction* of the air in passing through the fan nor that of the fan itself. The additional power required to offset these losses will be taken up later.

Example: The effective area of a fan outlet is 480 square inches, and the pressure within the casing is ½ ounce per square inch. What volume of air will be discharged per minute, and what H. P. will be required, neglecting friction?

From Table III. we find that for ½ ounce pressure, 25.37 cubic feet of air will be discharged per minute through an area of 1 square inch, at an expenditure of .00346 H. P. Therefore, the total quantity discharged will be $480 \times 25.37 = 12{,}177$ cubic feet, requiring $480 \times .00346 = 1.66$ H. P.

Relation between Volume, Pressure and Power.

It can be shown mathematically that the following relations are true in the case of an ideal fan, and tests have shown them to be approximately correct for fans in actual operation. (1) The volume of air delivered varies *directly* as the speed of the fan, that is, doubling the number of revolutions doubles the volume of air delivered. (2) The pressure varies as the *square* of the speed, for example, if the speed is doubled, the pressure is increased $2 \times 2 = 4$ times. (3) The power required to run a fan varies as the *cube* of the speed; again, if the speed is doubled the power required is increased $2 \times 2 \times 2 = 8$ times.

The value of a knowledge of these relations may be illustrated by the following example.

TABLE IV.

Temperature in Degrees F.	Volume for same Weight.	Relative Velocity due to the same Pressure.	Speed of Fan to Handle same Weight.	Speed to Produce same Pressure.	Power for Speed Required to Handle same Weight.	Power Required to Handle same Weight at same Pressure with a Properly Proportioned Fan.
1	2	3	4	5	6	7
50	1.00	1.00	1.00	1.00	1.00	1.00
100	1.10	1.05	1.10	1.05	1.21	1.10
150	1.20	1.09	1.20	1.09	1.43	1.20
200	1.29	1.14	1.29	1.14	1.67	1.29
250	1.39	1.18	1.39	1.18	1.93	1.39
300	1.49	1.22	1.49	1.22	2.22	1.49
350	1.59	1.26	1.59	1.26	2.51	1.59
400	1.68	1.30	1.68	1.30	2.84	1.68
450	1.78	1.34	1 78	1.34	3.18	1·78
500	1.88	1.37	1.88	1.37	3.56	1.88
550	1.98	1.41	1.98	1.41	3.92	1.98
600	2.08	1.43	2.08	1.44	4.32	2.08

The effect on the speed and power of a fan of increasing the temperature.

Suppose for any reason it was desired to double the volume of air delivered by a certain fan. At first thought we might decide to use the same fan and run it twice as fast; but when we come to consider that the power would have to be increased eight times, it is probable that it would be much cheaper in the end to use a larger fan and run it at a lower speed.

Effect of Temperature.

All computations and tables given thus far have been based on a temperature of 50 deg. F.

Raising the temperature of air causes it to expand and therefore reduces its density, or weight per unit of volume. This fact is of much importance where fans are used for induced draft as the temperature of the gases commonly ranges from 300 deg. to 600 deg.

Table IV. also from "Mechanical Draft" shows the effect on the speed and power of a fan when the temperature of the air is increased.

In the following example it is assumed for simplicity that the effective outlet area is equal to the blast area in each of the fans considered.

Example: From Table II. we see that a 4-foot fan running at a speed of 411 revolutions per minute, will produce a pressure of 1 ounce, and looking in Table III. we find that it will discharge 35.85 cubic feet of air per minute at a temperature of 50 deg. through an effective area of 1 square inch, with an expenditure of .00978 H. P.

If the width of the fan is 18 inches at the tips of the blades, the blast area may be taken as $\frac{48 \times 18}{3} = 288$ square inches, from which the delivery will be $35.85 \times 288 = 10{,}324$ cubic feet per minute, requiring $.00978 \times 288 = 2.8$ H. P.

Let us now assume the air to be heated to a temperature of 500 deg., and see what conditions are necessary to handle the same weight per minute. Looking in Table IV. for a temperature of 500 deg. we find that the volume becomes $10{,}324 \times 1.88 = 19{,}409$ cubic feet, and the speed of fan necessary to move this quantity is $411 \times 1.88 = 772$ revolutions per minute, requiring an expenditure of $2.8 \times 3.56 = 9.9$ H. P.

Suppose the above fan to be used for supplying a forced draft of 1 ounce to a battery of boilers and it is desired to change to induced draft where the gases are to pass through the fan at a temperature of 500 deg. What size and speed of fan will be required to produce the same intensity of draft (suction in this case) and what horse power will be required to run it?

It is evident that the weight of air required will be the same in each case. This, for forced draft, we found to be 10,324 cubic feet per minute at a temperature of 50 deg. and Table I. shows us that a peripheral velocity of 5,161.7 feet per minute was required to produce the pressure of 1 ounce at this temperature.

Referring to Table IV., columns 2 and 5, we find that after raising the air to a temperature of 500 deg. a volume of $10{,}324 \times 1.88 = 19{,}409$ cubic feet per minute is to be handled by the fan, requiring a peripheral velocity of $5{,}161.7 \times 1.37 = 7{,}071.5$ feet per minute to maintain the same pressure. As the velocity of flow through the discharge outlet is practically the same as that of the fan tips, the required blast area of fan will be $19{,}409 \div 7{,}071.5 = 2.74$ square feet $= 394$ square inches. Assuming a fan 60 inches in diameter, we have $394 \div \frac{60}{3} = 19.7$, or in round numbers, 20 inches as the required width at the periphery. This gives a fan of very nearly the same proportions as the 4-foot fan first used. The circumference of a 6-foot fan is 18.8 feet, therefore, 7,071.5, the required peripheral velocity, divided by $18.8 = 376$ revolutions per minute, the required speed of the fan.

The H. P. required by the 4-foot fan in handling the same weight of air at 50 deg. was 2.8.

Referring to Table IV, column 7, we find that the power required to deliver the same weight of air at the same pressure at a temperature of 500 deg. is 1.88 times as great, or $2.8 \times 1.88 = 5.2$ H. P.

* * *

The terrible boiler explosion at Brockton, Massachusetts, last March, which caused the death of 58 people, incites the *Mechanical Engineer* to comment in no uncertain way on the alleged defects of boiler making in the United States. The editor says, that "American boiler making is twenty years behind that of Great Britain. Flat plates, imperfect bending, hand riveting and lap-joints for small diameters and high pressures are to-day common features of the American boiler factories, which largely contribute to the intensely high rate of boiler mortality." In short, lap joint and longitudinal seam are features of boiler construction which would not be tolerated in Great Britain; but unfortunately by far the greater percentage of American boilers are made with this defective joint, which may develop hidden cracks that are not discoverable by any ordinary methods of inspection.

* * *

It is said that the Niagara River develops 8½ million H. P. continuously. Such an amount of energy is of course beyond our comprehension, but some idea may be obtained by considering the amount of coal which would have to be consumed to develop energy at this rate. Assuming 2 pounds of coal burned per H. P. per hour, the weight consumed in an hour to equal the work of Niagara River would be 8,500 tons. Continuous work at this rate for one year would require 74,460,000 tons of coal, which is more anthracite than the United States produced in the year 1900.

VARIABLE SPEED MECHANISMS.—5.

RATCHET DEVICES.

The most common variable speed mechanism used on machine tools, aside from the familiar step cone pulleys, is the feed motion, generally employed on planers, slotters and shapers, which utilizes some form of the familiar pawl and ratchet. By varying the throw of the pawl the amount of angular motion is varied and, of course, the velocity, although such devices are not usually regarded as speed variators in the sense here employed. The principle, however, has been more or

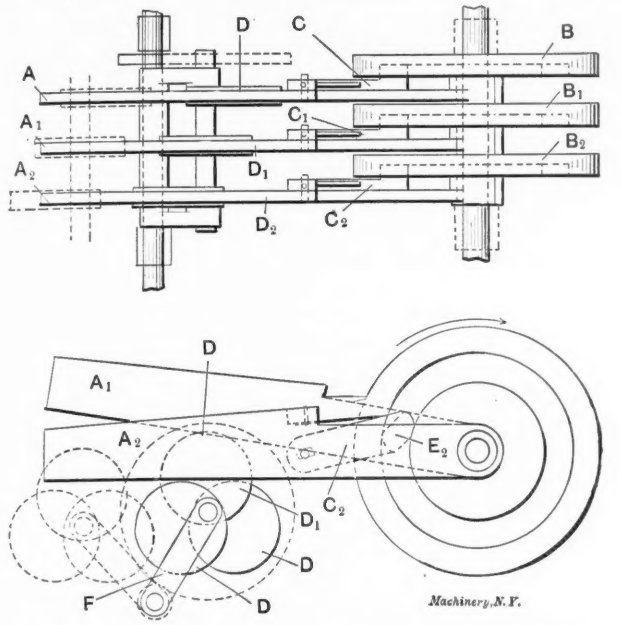


Fig. 48. Patent No. 297,143, granted to W. F. Marks, April 22, 1884.

less successfully utilized in speed varying mechanisms by providing a multiplicity of ratchets and pawls, so that a continued action is maintained instead of an intermittent action as is the case with only one pawl and ratchet.

Fig. 48 shows two drawings accompanying patent specification No. 297,143, granted to W. F. Marks, April 22, 1884, for a variable speed device based on the ratchet principle. It includes the three levers, $A\ A_1\ A_2$, which operate the grooved disks, $B\ B_1\ B_2$ by means of clutches. The clutches are fastened to the sides of the levers and are in the form of three pawls, $C\ C_1\ C_2$, which have projecting lugs of the shape shown

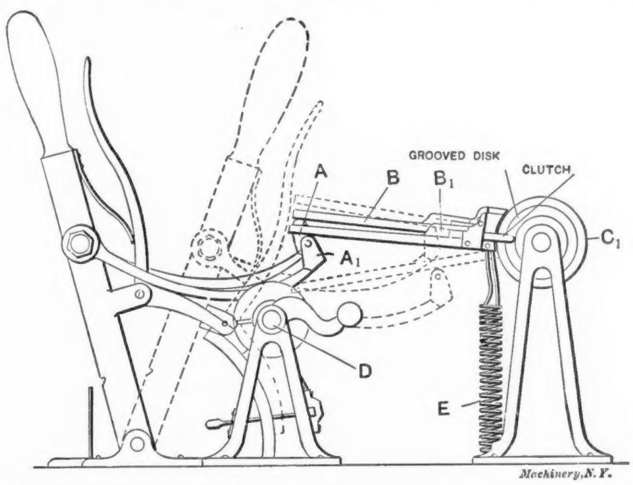


Fig. 49. G. H. Preston's Device for Changing Speed. Patented June 17, 1884. No. 300,734.

in the outline, E_2, in the lower view. These lugs engage the side of the grooves in the disk when the levers are lifted, and thus impart motion; but on the downward movement the gripping action is released, the same as with an ordinary pawl and tooth ratchet. In short, the inventor has provided a friction ratchet whereby he gains the advantage of having an infinite number of teeth and is thus enabled to provide increments of any degree. The levers are operated by eccentrics, $D\ D_1\ D_2$, mounted on a shaft which is in turn mounted on swinging arms so that the position of the eccentrics relative to the levers may be changed at will. It is obvious that when the eccentrics are in the position shown by the full lines in the upper view, the levers are given a greater angular movement than when in the position indicated by the dotted lines. Hence, the variation in speed is obtained by swinging F within the limits indicated.

Patent No. 364,530 was granted to A. J. Martin, June 7, 1887, for the changeable speed gearing illustrated in Fig. 50. This device employs two crank wheels, $A\ A_1$, having cranks of variable throw, each crank pin being connected to a rack which meshes with a pinion at the center of the crank disk. The connecting-rods, $B\ B_1$, are attached to sliding cross heads which communicate the reciprocating motion to the toggle-joint arms, $C\ C_1$. The inner ends of each set of toggle-arms are pivoted on the center of the grooved disk, $D\ D_1$, and engaging these disks are clutches attached to the arms by which motion is imparted to the disks. These clutches grip the sides of the groove with one direction of motion of the crank wheel, as for example in a clockwise direction, and release in the opposite direction. Suppose that the direction of motion of the upper arm, c_1, is in the same direction as the crank wheel and that its clutch engages the groove in disk D_1: It follows that the disk D will have circular motion imparted to it in the direction of the arrow. When the crank reaches the end

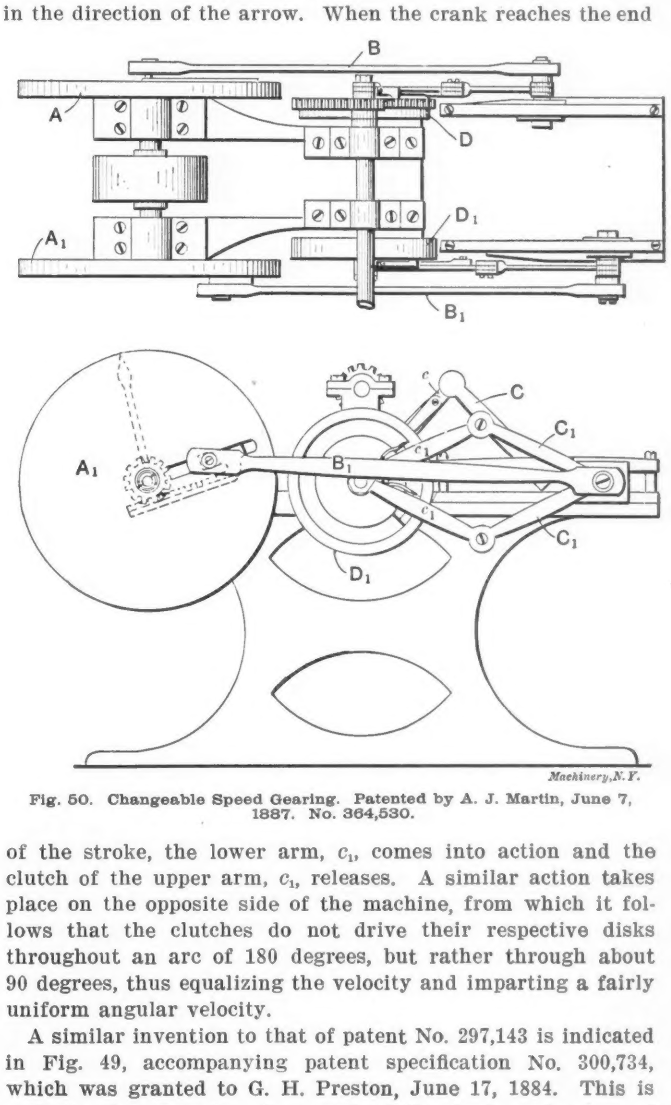


Fig. 50. Changeable Speed Gearing. Patented by A. J. Martin, June 7, 1887. No. 364,530.

of the stroke, the lower arm, c_1, comes into action and the clutch of the upper arm, c_1, releases. A similar action takes place on the opposite side of the machine, from which it follows that the clutches do not drive their respective disks throughout an arc of 180 degrees, but rather through about 90 degrees, thus equalizing the velocity and imparting a fairly uniform angular velocity.

A similar invention to that of patent No. 297,143 is indicated in Fig. 49, accompanying patent specification No. 300,734, which was granted to G. H. Preston, June 17, 1884. This is a device for changing the speed of the feed mechanism of grain drills. The principal novelty of this device, as compared with that of 297,143, is in the construction of the clutch mechanism, and that cams are used in place of eccentrics. The cams have a harmonic motion for a little more than one-half their circumference and this feature permits a fairly uniform movement to be obtained with only two ratchet mechanisms instead of three as in No. 297,143. Referring to the cut, Fig. 49, D is the cam shaft on which rest the rocking levers, $A\ A_1$. The rotation of the cam shaft imparts motion to the clutch levers, $B\ B_1$, which actuate the grooved disks, one of which is shown at C_1. With the levers in the position shown by the full lines, the angular movement imparted to the disks

per revolution of the cam shaft is the minimum; and the maximum angular displacement is obtained when the levers are in the position indicated by the dotted lines. The springs shown at E keep the clutch levers and the cam levers in close

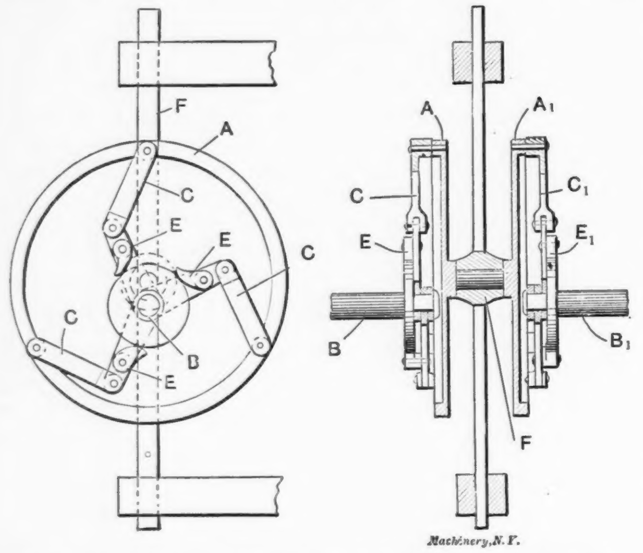


Fig. 51. Patent No. 634,327, granted to L. M. Dieterich, October 3, 1899.

contact with the cams at all points of rotation. The pawl of the friction clutch is a hook-shaped member which spans the rim of the grooved disk, gripping it on the upward movement and releasing it on the downward movement.

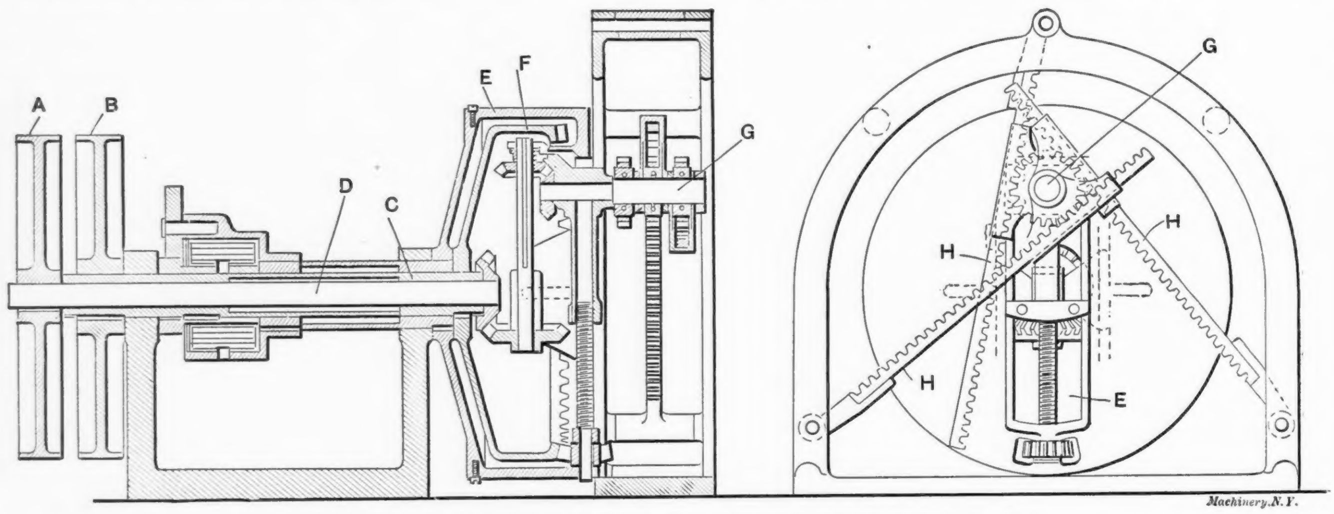


Fig. 52. R. P. B. Green's Changeable Speed Gear. Patent No. 685,834, November 5, 1901.

The variable power transmitter illustrated in Fig. 51 was patented by L. M. Dieterich, October 3, 1899, No. 634,327. In this device variation in the relative angular velocity of shafts $B\,B_1$ is obtained by a lateral displacement of the central member, F, which forms the bearing support of the disks, $A\,A_1$. Supposing that B is the driving shaft, it transmits motion to A through the medium of the friction pawls E which are pivoted on the jointed levers C. When A and shaft B are concentric it is evident that each pawl will act as a driver throughout a rotation; but when F carrying A and A_1 is displaced laterally, it then follows that only one of these pawls will be driven at a time, and through about one-third of the arc of a circle only. During the remaining two-thirds it will travel slower than the disk and so fall behind, since the outer ends of the links $C\,C$ are further removed from the center of the shaft B during two-thirds of its rotation. The converse of this action takes place on the opposite side with the disk A_1, levers C_1, and pawls E_1, which transmit the motion to the shaft B_1. From this condition it follows that B_1 is driven at the same rate as B when the disks A and A_1 are concentric with the driving and driven shafts; but when they are displaced sideways, the driven shaft runs at a faster rate than the driver, depending upon the displacement.

In the light of these devices previously described, the principal novelty of the patent 637,477, granted to W. H. Newman, November 21, 1899, for a variable speed gear, lies in the method employed for speed control. Referring to Fig. 53, there are two eccentrics, both of which are of the compound type, as shown in the upper view at $D_1\,E_1$; that is, each is composed of two eccentrics, one mounted upon the other and both supported by shaft F. In the position shown the compound eccentric has its maximum throw, but shifting D_1 180 degrees so transforms the eccentric that it has no throw; hence no motion will be transmitted to the rocking levers.

Patent No. 685,834 was granted to R. P. B. Green November 5, 1901, for a variable speed gear of complicated, not to say defective construction, but interesting principle. One drawing of this specification is shown in Fig. 52. The pulley B is supposed to be the driving pulley, transmitting motion through the sleeve C to the hollow crank-wheel E. On the crankpin G are mounted three pinions having ball clutches, and meshing with these pinions are three racks, H. As the crank is rotated the pinions receive motion from the racks of varying velocity, depending upon the relative position of the crankpin and the racks. The pinion which receives the fastest motion grips the pin G, which also acts as a rotary member, and causes it to drive the shaft D through the train of bevel gears, thus making A the driven pulley. The bevel wheel F and connecting mechanism appear to be for shifting the crankpin G radially on E and thus change the speed rate. Another drawing in the same specification shows the motion taken from the pin G by means of a shaft containing universal joints.

Fig. 54 shows, diagrammatically and constructionally, a bicycle variable speed gear that was patented by A. Sharp, January 28, 1902, No. 692,077. The rear sprocket, A, may be shifted laterally with reference to the wheel, thus locating it either in a concentric or eccentric position thereto. In the diagram it is shown eccentric to the rear wheel hub in which position it should drive the rear wheel at a higher ratio than when

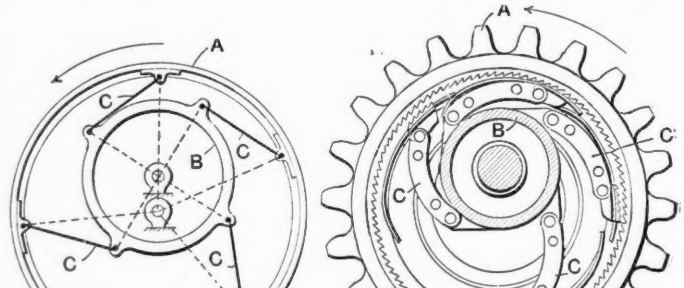


Fig. 54. Patent No. 692,077, granted to A. Sharp, January 28, 1902.

concentric. The inner face of the sprocket is provided with ratchet teeth for engagement with shoes having similar shaped teeth that are attached to the ends of the pivoted connecting members C. These are attached to the wheel hub B. The

side of the sprocket which is nearer the center of the wheel hub becomes the driving side; hence the angular velocity is greatest with a maximum displacement of the sprocket relative to the wheel hub.

A patent, No. 700,970, was granted May 27, 1902, to J. D.

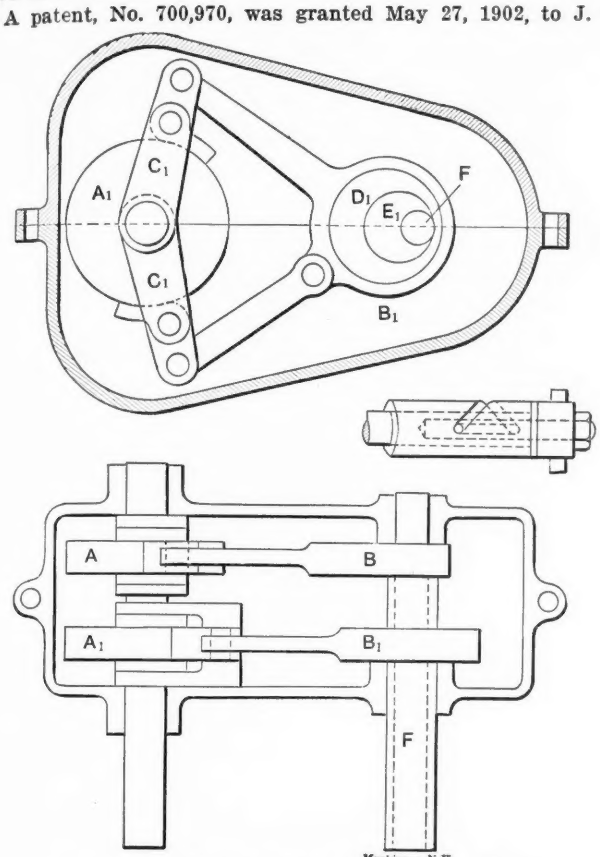


Fig. 53. Variable Speed Gear, Patented by W. H. Newman, November 21, 1899. No. 637,477.

McFarland, Jr., for an adjustable speed gear designed for the use of automobiles. It consists essentially of a shaft, A, having helical grooves or tongues, and a sleeve, B, sliding thereon. The helical tongues, as shown in Fig. 55, are made of varying pitch, the pitch changing from a straight line parallel to

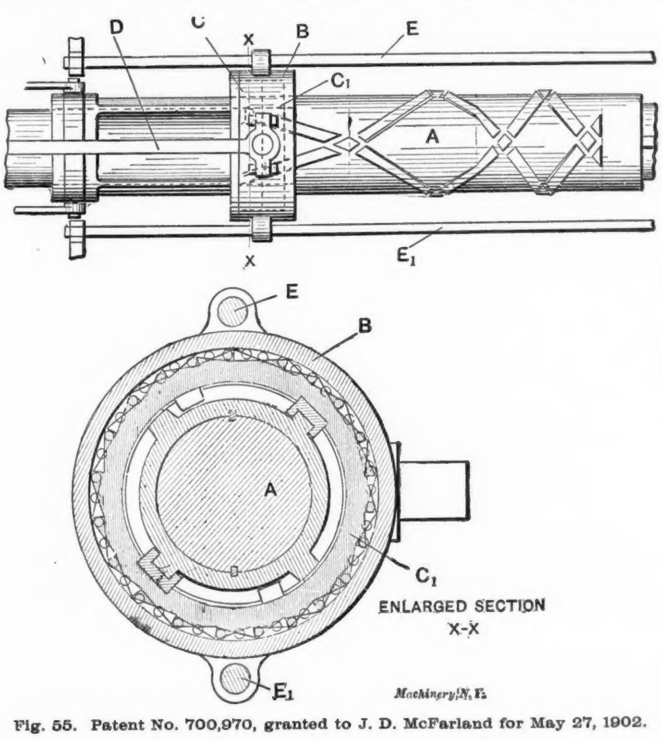


Fig. 55. Patent No. 700,970, granted to J. D. McFarland for May 27, 1902.

the axis to the maximum, by increments of say 10, 20, 30, 40 and 50 degrees. The lines of the tongues are continuous, however, and the pitch of each section is uniform. Both right and left-hand helices are provided, with which are engaged the elements of right- and left-hand nuts; the elements of one nut being secured to the inner ring C and the other to ring C_1. These rings have ratchet seats cut in their peripheries in which are placed balls, thus forming ball clutches. One ratchet is pitched in a right-hand direction and the other left-hand. Now, when reciprocating motion is communicated to sleeve B by means of the connecting-rod D, the guide rods E E_1 prevent the sleeve B turning with the helices on A, thus constraining the shaft A to rotate, the direction depending, of course, upon the relative arrangement of the ratchets and the

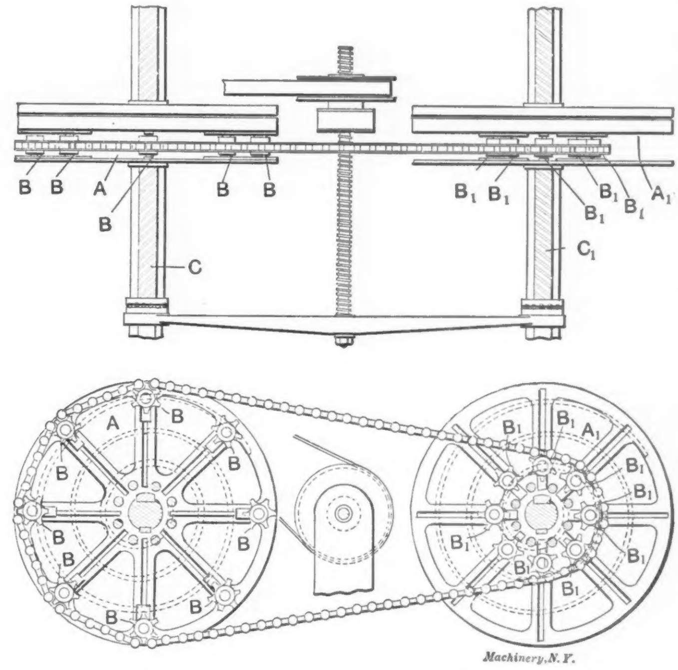


Fig. 56. Patented by W. N. Dumaresq, April 7, 1903. No. 724,450.

helical tongues. The reverse movement of the directing rod continues the motion in the same direction inasmuch as the other ratchet comes into action and transmits motion to the helix of opposite pitch. The position of sleeve B upon A determines the speed ratio, the ratio being greatest on the helix of greatest pitch.

The variable speed gear shown in Fig. 56 was patented by W. N. Dumaresq, April 7, 1903, No. 724,450. In this device the driving and the driven members, A A_1, consist of slotted disks carrying small sprockets. The drawing shows the use of sprockets and chain similar to that used on bicycles. The

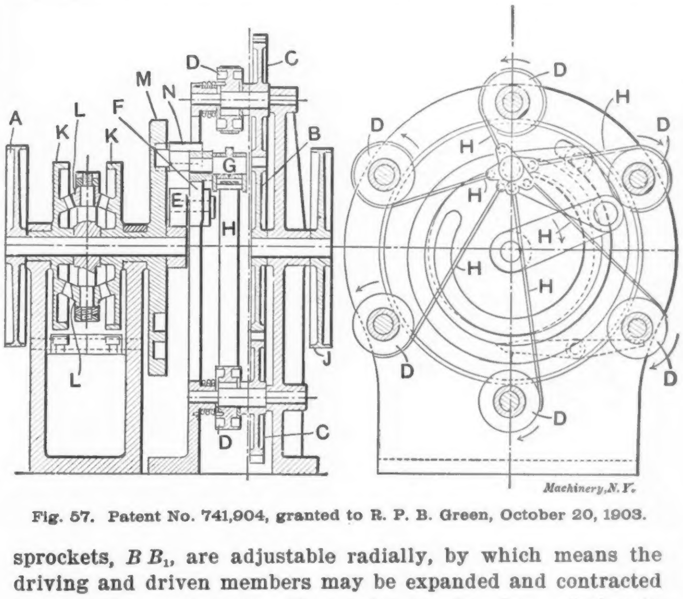


Fig. 57. Patent No. 741,904, granted to R. P. B. Green, October 20, 1903.

sprockets, B B_1, are adjustable radially, by which means the driving and driven members may be expanded and contracted to vary the speed ratio. To compensate for the variation in pitch the sprockets are mounted on roller clutches so that each sprocket is free to adjust itself to the correct pitch as the wheels rotate. The adjustment, of course, takes place by each sprocket turning in a forward direction, as backward movement is prevented by the locking of the clutches. The mechanism by which sprockets are shifted radially consists

essentially of a scroll plate with which lugs on the sprocket mountings are engaged. These scroll plates are rotated by shifting an internal coarse-pitch screw, $C\,C_1$, longitudinally relative to the scroll plates, causing same to revolve independently of the slotted disk and thus change the position of the sprockets. This movement, of course, takes place simultaneously on both the driving and driven wheels and in opposite directions so that an even tension is maintained in the driving chain. It is apparent that this device is one form of expanding pulley adapted to the use of chain, thus making it positively driven by use of the roller clutch sprockets.

An improvement on patent No. 685,834 granted to R. P. B. Green, November 5, 1901, was granted to the same patentee October 20, 1903, No. 741,904. The power is transmitted to this device by means of pulley *A*, Fig. 57, which is mounted on a shaft carrying the crank *E* on its inner end. To this crank is pivoted a link, *F*, to which is attached the crankpin *G*. To the crankpin are connected a number of belts, *H*, which wind upon pulleys, *D*. These pulleys are retracted by coil springs so that the belts are kept under constant tension. As the crank wheel is revolved motion is communicated to pulleys *D* at varying velocities, the velocity of each, of course, depending upon the relative position of the crankpin and each pulley in turn. The coil springs also serve to engage toothed clutches formed on the side of each pulley and the adjacent side of the pinions *C*. As each pulley rotates at maximum velocity it acts as a driver for its mating pinion, which motion is transmitted to the gear *B* and thence to the driven pulley *J*. The device suggested in the cut for varying the speed consists of a differential gear. The gears $K\,K_1$ are provided with smooth peripheries for brakebands, the hub on which pinions *L* are mounted is keyed to the shaft and normally the combination rotates with it. Should it be desired, however, to change the position of crankpin *G* in order to vary the velocity ratio, a brake is applied to either *K* or K_1, thus retarding its motion and causing the snail cam *M* to move faster or slower than *A*, the consequence being that the crankpin is moved in or out relative to the center, thus changing the speed ratio. In none of these devices does the ratchet mechanism have control of the driven member if there is any tendency to overrun.

Fig. 1. Birds-eye View of B. F. Sturtevant Plant. Machine Shop is the Central Building with Saw-tooth Roof.

Fig. 2. Interior of Machine Shop. This Shop is Constructed on the Gallery System, with Wide Central Bay, but differs from other Buildings of this Type in that a Saw-tooth Roof is used for Lighting the Central Bay and the Inner Sections of the Galleries.

THE MANUFACTURING BUILDINGS OF THE B. F. STURTEVANT CO.

In November, 1902, we published a general outline, with a plan, of the new works of the B. F. Sturtevant Co., that were then under construction at Hyde Park, Mass. The old plant at Jamaica Plain, Mass., had become outgrown, there was no opportunity for growth at the old location and it was finally decided to move to a locality where ample acreage could be obtained, and erect new buildings throughout. While negotiations were in progress for a site the Jamaica Plain plant was visited by a severe fire, which made it imperative to hasten the construction of the new plant as rapidly as possible. A tract of twenty acres was purchased at Hyde Park, Mass., and the new buildings were laid out on an extensive scale, having a floor area more than double that of the old plant, and actually aggregating nine acres.

The disposition of the buildings was determined by the provision to be made for growth and is clearly indicated by the birds-eye view in Fig. 1. The arrangement provided for a group parallel to the railroad tracks of the N. Y., N. H. and H. railroad with accommodations for spur tracks between the buildings, their entrance at the ends of certain buildings, and an opportunity for growth of all important structures by extension in length. At the front is the office building, and directly back of it a building devoted to engine testing and shipping and to the manufacture of generators, motors and electrical apparatus. Extending backward from this latter building are two structures 500 feet long, the further one in the engraving, with the saw-tooth roof, being the machine shop and the nearer one the fan shop, where also some electrical work is done. Beyond, to the left, are the foundry, pattern shop, pattern storage, etc., and at the extreme rear of the works is the power plant. The smith shop is between the machine shop and fan building.

A description of the foundry and pattern buildings was published in MACHINERY in November, 1903, and we are now able to publish photographs of the machine and manufacturing departments, which give a clear idea of the design of the buildings, and of their equipment.

Features of Construction.

The type of construction of these buildings is somewhat composite in its character. They have steel interior columns and main steel girders, with heavy brick walls, wood timbered floors and plank roofs. In the case of the one-story foundry, the roof is supported by steel trusses. In the other buildings open timbering with wooden columns in the upper floor is employed. The main floors in the machine, fan and erecting shops are of tar concrete, upon which 3-inch hemlock is bedded in liquid pitch and toe-nailed together. The upper floors are carried upon hard pine beams on 4-foot centers spanning the spaces between the steel girders, which follow a unit system of 20 feet on centers throughout the buildings. The machine shop gallery floors which are designed for 250 pounds load per square foot, are of 2½-inch plank; other upper floors are 2-inch plank for 200 pounds per square foot. Maple top flooring is used in all cases. All roofs are of 3-inch plank with tar and gravel top.

The standard first floor height in the main buildings is 17 feet, that of the second and third stories is 15 feet. The windows are large and numerous. Ribbed glass is used in all but the lower sashes. All structural steel, window frames, racks, and bins and the walls up to a height of 5 feet are painted a rich green. The balance of walls and ceilings is covered with white cold water paint.

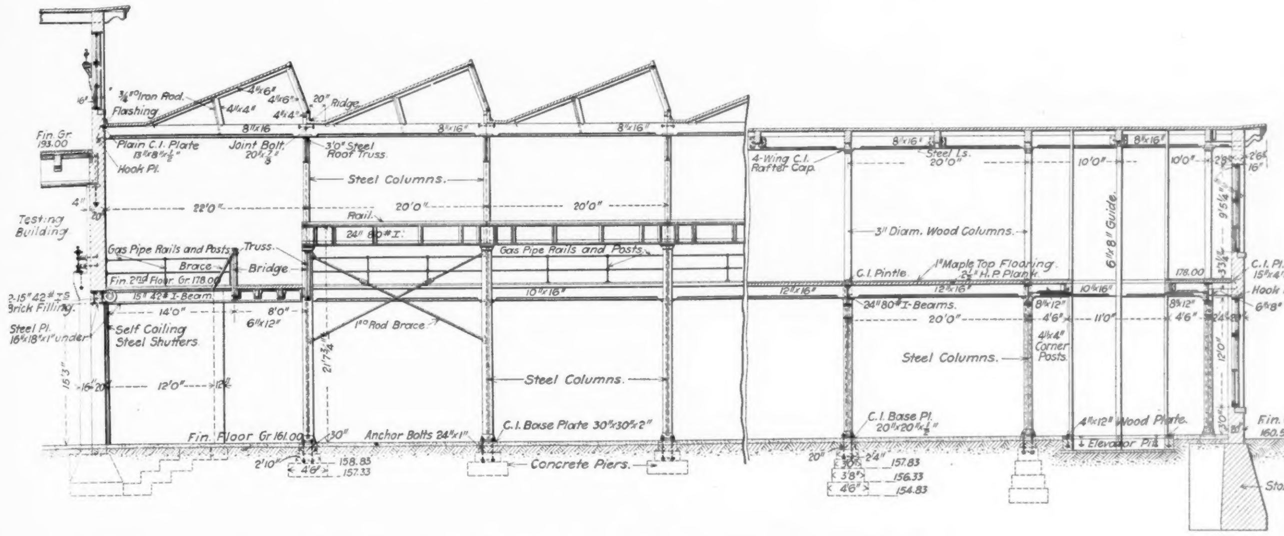


Fig. 3. Longitudinal Sections of the Central Portion of the Machine Shop, showing Saw-tooth Roof, and also of one of the Gallery Sections.

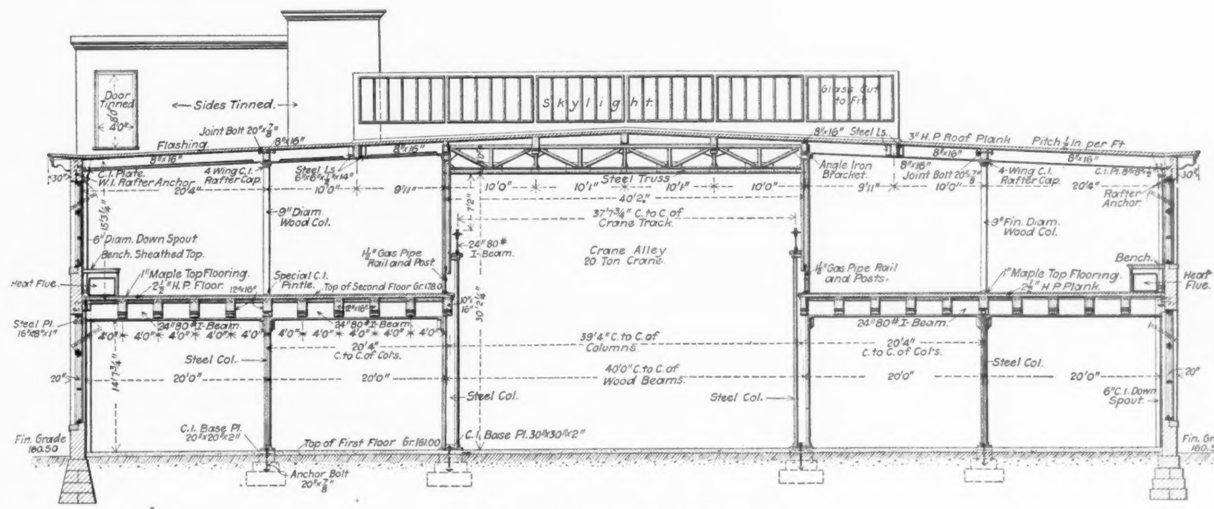


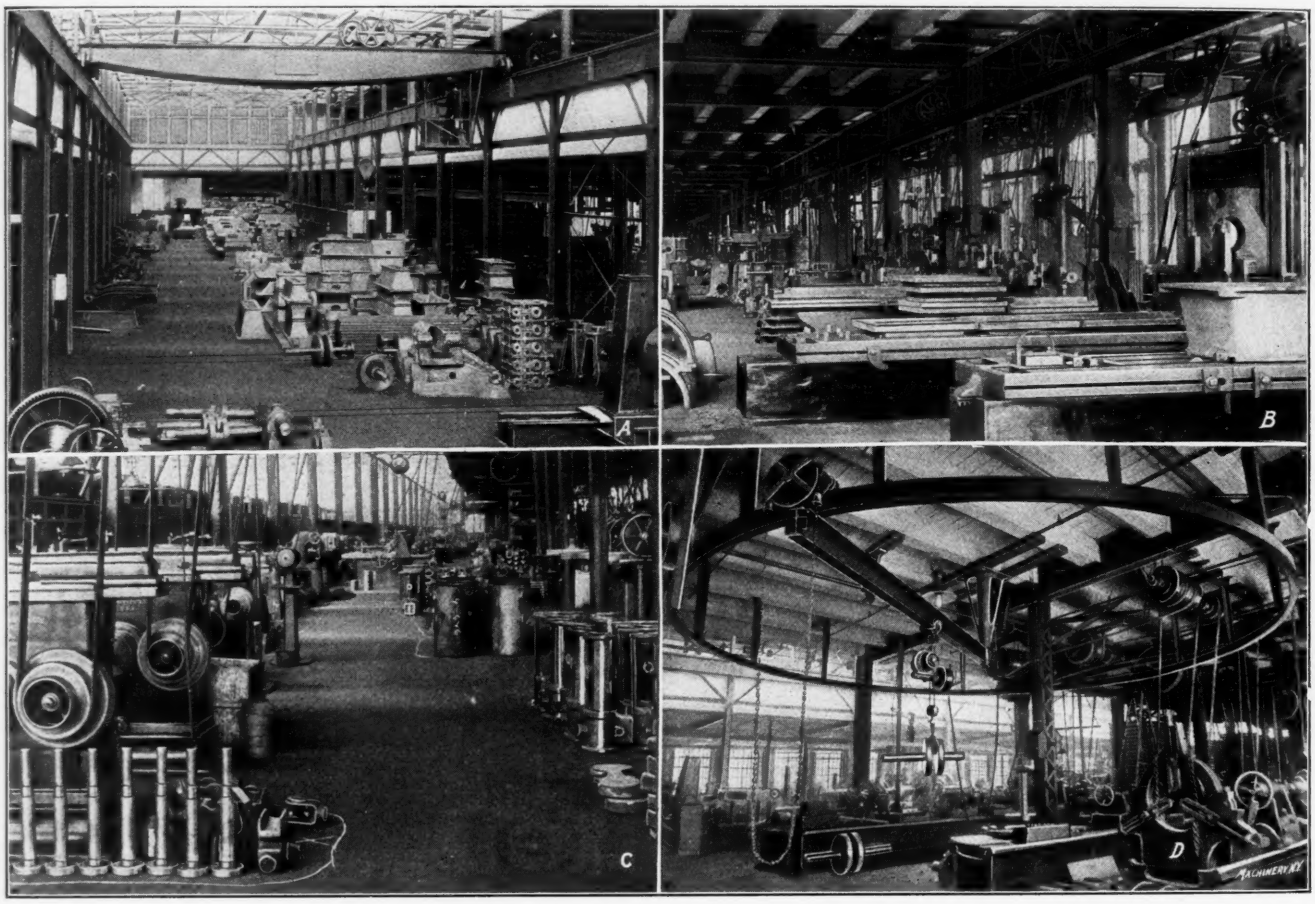
Fig. 4. Cross-section of Machine Shop.

The Machine Shop.

The machine shop is of the familiar gallery type, with wings 40 feet wide, and central craneway of the same width designed for crane of 20-ton capacity; making a total width of 120 feet. The lighting, which is remarkably effective, is secured principally by a series of sawtooth skylights running crosswise of the roof with glass facing due north.

The present crane of only 10 tons capacity serves the entire floor and finally deposits the substantially complete engine or generator upon a transfer car which passes through to the testing building where a 15-ton crane picks up the machine, drops it upon the testing plate and subsequently carries it forward to the steam railway track which passes through the end of the building and provides space for the loading of two cars at a time.

Upon the ground floor in the corner nearest the erecting shop is located the general stores room and the office of the stores keeper. Here are all general stores, brass and malleable castings and the like, as well as completed parts of engines, ready for assembling.



A.—General View in Shop. B.—View under one of the Galleries. C.—View on one of the Galleries, showing Lighting from above. D.—Circular Crane serving three Tools.

FIG. 5. GROUP SHOWING CERTAIN FEATURES OF CONSTRUCTION.

Tools are arranged for the progress of the work from the open end of the shop toward the finished stores and erecting shop. Until such time as an independent department is provided for the manufacture of economizers, equipment for this work is provided near the entrance to the machine shop. Otherwise the grouping of machines is in accordance with the general scheme.

The second floor is served by four elevators and an equal number of stairways, one at each end and two at the center. The galleries are connected at the ends by bridges; the grinding and polishing room being located at the end adjacent to the erecting shop. Close beside it on the same floor is the brass finishing department and the office of the superintendent of the machine department. In the opposite gallery in the most accessible position are the tool making and tool storage rooms. An enclosed bridge connects with the second floor of the fan shop.

Machine tools are grouped by types, and so far as possible arranged for progressive operations. Space is left for additions in supplementary parallel rows. Large machines are driven by direct attached Sturtevant motors. Twenty horsepower motors suspended overhead are so located as to drive the small tools in groups of suitable size through individual lengths of line shaft.

The Fan Building.

The building devoted to the manufacture of fans, heaters, etc., is 80 feet in width, of the same length as the machine shop, and three stories in height, of typical mill construction, provided with all conveniences for handling material and arranged so that goods can be received at, and shipments made from, numerous points along one side, while other supplies are brought in from the court between it and the machine shop.

One half of the length of the fan shop is served by a 5-ton traveling crane, the floor to floor height throughout the area being 32 feet to provide for the construction of large steel plate fans for ventilation, mechanical draft and the like. Here also are built the heater jackets, some of them large enough for a summer cottage. In the adjoining space on the first floor are installed large shears and brakes for cutting and folding plates up to 120 inches in length and rolls for ½ inch plates 120 inches wide. Pneumatic punches, riveters and chippers are extensively used in this building. Plate iron is completely stored on edge in diagonal alcoves in the storage shed alongside the fan shop, which has capacity for nearly a thousand tons.

Midlength of the first floor is the packing and shipping department, near the large five-ton elevator which serves all floors. The other end of this floor is devoted solely to the manufacturing of heater sections for the Sturtevant steam hot blast apparatus.

On the second floor smaller fans, forges, etc., are built and half of the third floor is devoted to galvanized iron work. Part of this floor also is devoted to the punching and commutator division of the electrical department.

Testing Building.

The upper floors of this building are given up to the electrical department. In one end of the latter space occupied by

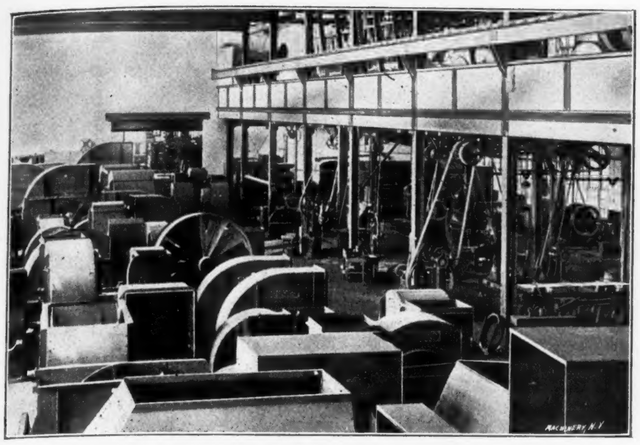


Fig. 6. Erecting Floor for Fans and Heater Jackets.

Fig. 7. Testing Electrical Fans in Testing Department.

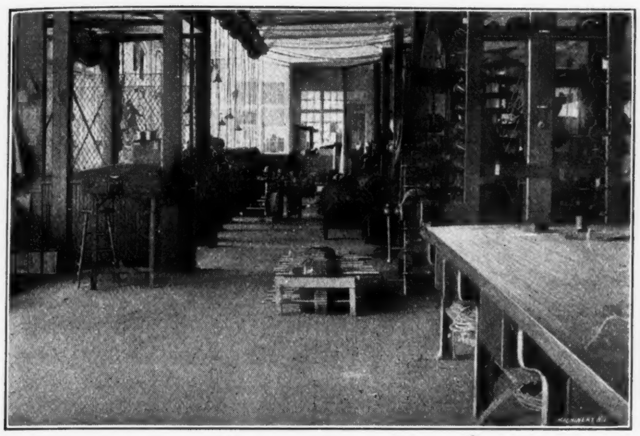


Fig. 8. A Portion of the Winding Division in Electrical Department.

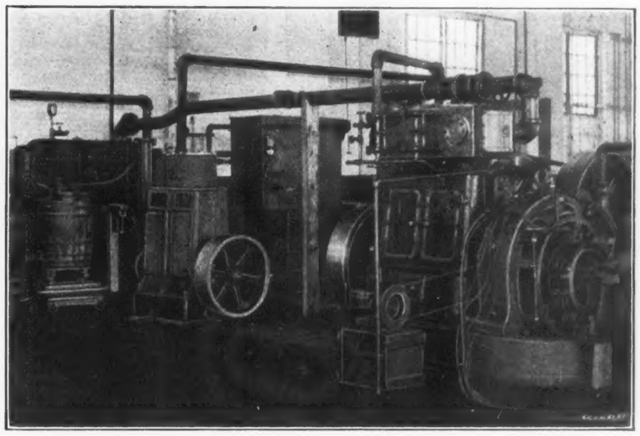


Fig. 9. Engines under Test on the Testing Plate.

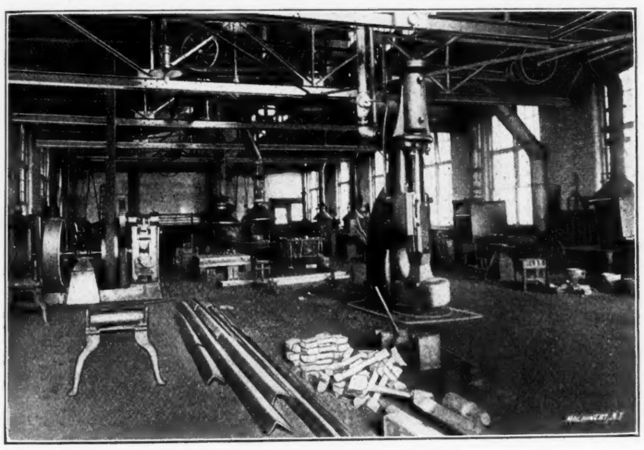


Fig. 10. Interior of Forge Shop.

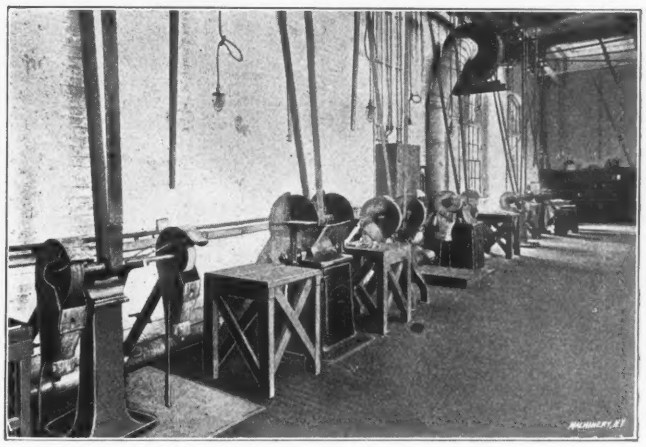


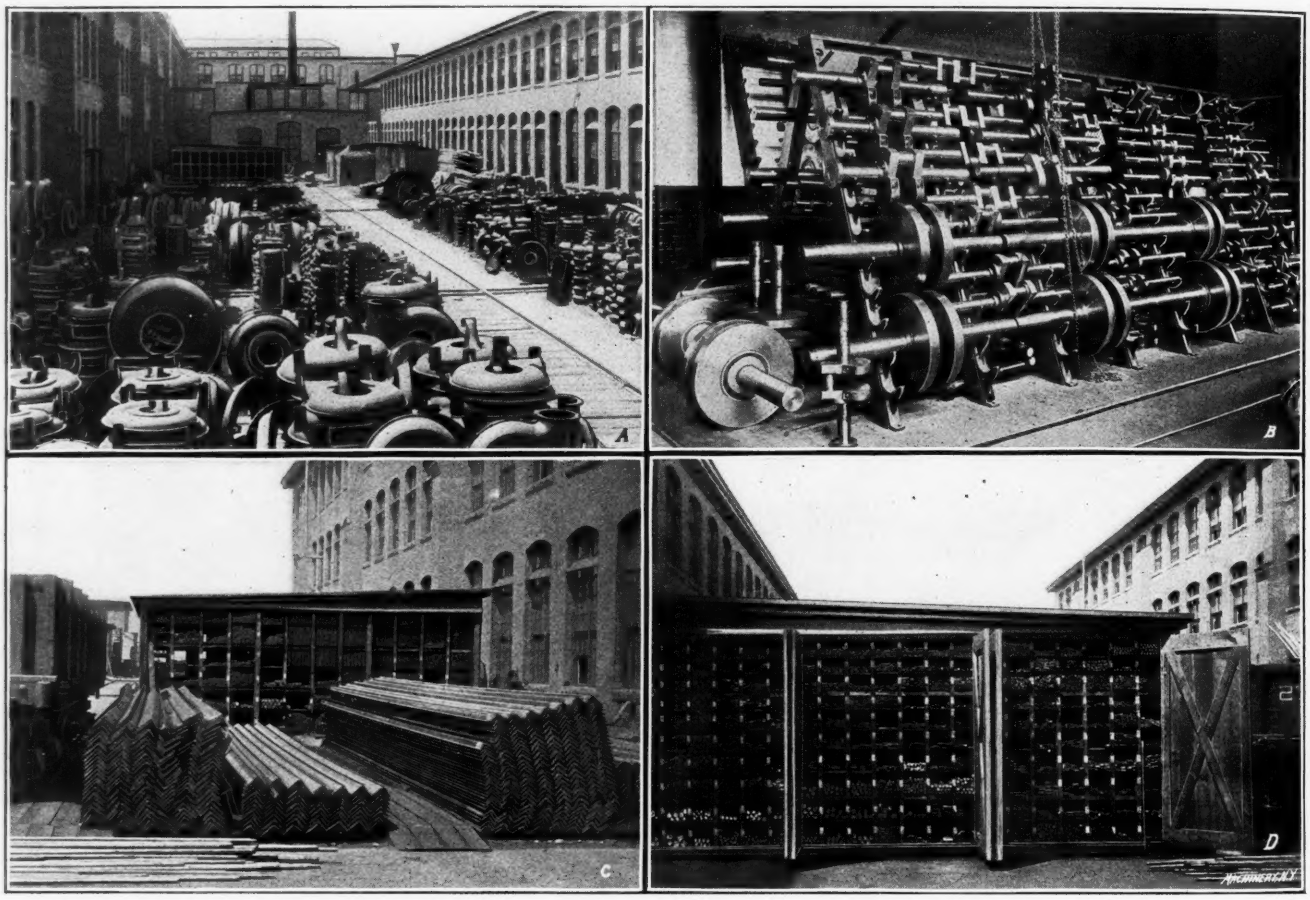
Fig. 11. Grinding and Polishing Room, with Exhausting Fan.

Fig. 12. Second Floor. Wash and Locker Building.

Fig. 13. Emergency Room.

VIEWS OF SEVERAL DEPARTMENTS OF WORKS.

this department is the baking room for armatures, etc. This measures 40 feet square, is entirely fireproof, and contains two steam heated ovens. At the other end of the building is the special store room for electrical supplies. The balance of this floor and of the intermediate floor below is devoted to winding, assembling, testing, etc. Fans are here equipped with motors and run under test conditions. Generators for direct connection are carried down to the first floor of the same building where they are attached to their respective engines and continuously run upon the test plate. The balance of the first floor is given to the assembling of engines and to the packing, storage and shipment of these machines. The testing plate measuring about 30 feet by 60 feet is completely equipped with steam and electrical connections; engines may be run condensing or non-condensing and efficiency tests conducted. This very complete equipment has proved invaluable in conducting the rigid tests demanded by the U. S. Navy Department upon the engines, generating sets, engine and motor-driven fans, which this company furnishes the government.



A.—Castings Storage between Buildings. This Area is Served by the Industrial Railway. B.—Storage of Crank Shafts in Machine Shop. C.—Piling Floor and Rack for Structural Work. D.—Storage Rack for Bar Iron, Steel, Etc.

FIG. 14. GROUP SHOWING ARRANGEMENTS FOR THE STORAGE OF STOCK.

All of the buildings are heated and ventilated by the Sturtevant System. In the machine shop the hot air pipes are hidden beneath the second floor wall benches and deliver most of the air downward to the first floor. The fan and erecting shops are supplied through an underground duct which delivers the hot air to external vertical flues upon one side of the building, which in the case of the fan shop, are located 40 feet on centers and discharge the air across the building above head level. The engines, generators, and some of the other apparatus of the power plant are also of Sturtevant manufacture.

* * *

The surprising variety which exists in the applications of hot blast drying apparatus is exemplified by recent installations undertaken by a well known firm engaged in this business. Among these are comprised installations for drying hops in the Sacramento Valley, lumber in Iowa, cloth in North Carolina, brick in Georgia, gun powder in New Jersey, lace curtains and handkerchiefs in Illinois, fish fertilizer in New York, rubber in Massachusetts, artificial leather in New Jersey, and plaster in New York.

POWER TRANSMISSION CHART.

R. N. DULL.

The accompanying chart is for determining the several quantities involved in the transmission of power by belts and by paper frictions. The chart is based on good practice and is the most convenient of the kind that I have seen. It takes into account the revolutions per minute of pulleys, the surface speed, the diameter in inches, the horse power, the width of face of a pulley, and the kind of belt used. The vertical lines at the right end with abbreviations at the top which represent different ply rubber belting, double, light double, and single ply leather belting and paper frictions. The coefficient of friction is assumed to be 0.3, with an arc of contact of 135 degrees for the belting, and a coefficient of 0.2 for the paper friction. The maximum working stress on leather was assumed at 250 pounds per square inch and 11 pounds per ply one inch wide for the rubber; while a pressure on the paper frictions was taken at 150 pounds.

The above assumptions are based on good and conservative practice and the horse power may be increased 25 per cent for the amounts determined on this basis, in extreme cases, although this increase should be avoided if possible.

The chart can be used to find the following:

Surface Speeds "S."

Intersect the vertical D with the horizontal N.

To Find Face "W."

Intersect the vertical D with the horizontal N; thence, diagonally upward to the left, or downward to the right, to the vertical H; thence, horizontally to the vertical lines at the right end of the chart; thence, diagonally upward to the right to W.

To Find Diameter and Face for Given Power and Speed.

Intersect the vertical H with the horizontal N; thence, diagonally downward to the right, or upward to the left, to intersect with the vertical D for trial; thence, horizontally to the vertical belt line; thence, diagonally upward to the right to W.

To Find Horse Power for Given Wheel.

From W trace diagonally downward to the left to the belt line; thence, horizontally to the vertical D; thence, diagonally downward to the left to intersect the horizontal N; thence, vertically downward to H.

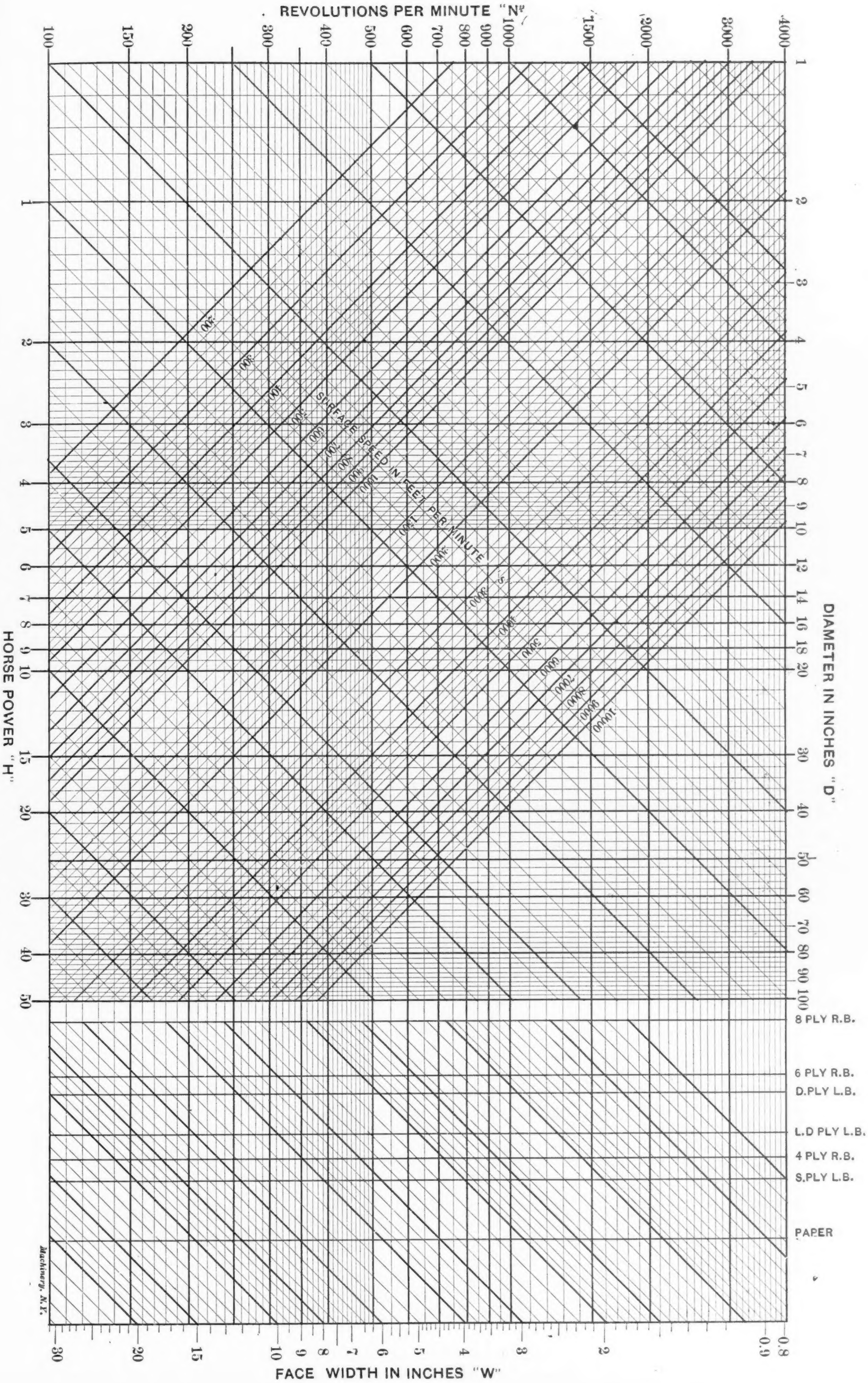


Chart for Determining Power Transmitted by Belts and Paper Friction Wheels.

THE FRICTION OF ROLLER BEARINGS.

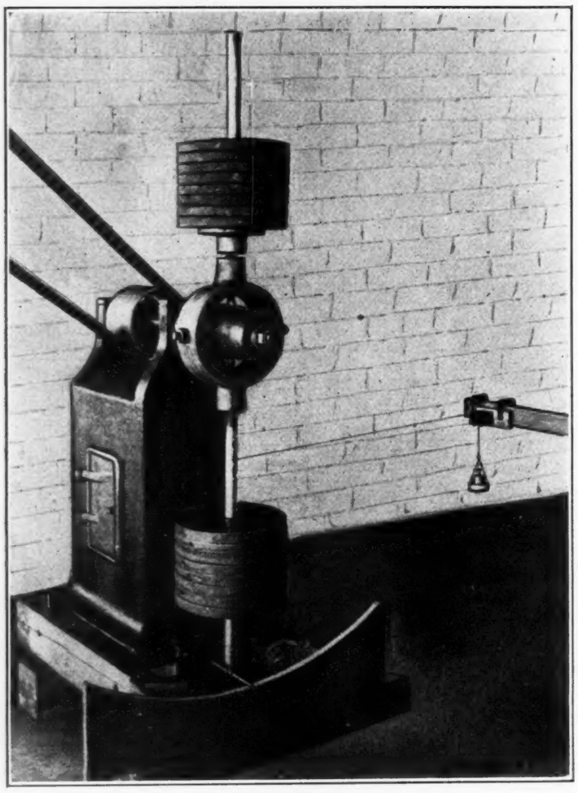
C. H. BENJAMIN.

During the years 1904-05 a series of tests on roller bearings was conducted under the writer's direction at the Case School by senior students, Messrs. Howells, Thomas, Woodhouse, Jandus and Collister.

It is to be supposed that every one knows the friction of roller bearings to be less than that of plain bearings, but it is doubtful if many people know how much less.

An attempt was made in these experiments to compare roller bearings with plain cast-iron bearings and with babbitted bearings under similar conditions. Four sizes of bearings were used in the tests, measuring respectively 1 15-16, 2 3-16, 2 7-16 and 2 15-16 inches in diameter. The lengths of journals were about four times the diameters.

The bearings were in two parts and were held in a circular yoke by setscrews. This yoke carried two vertical spindles, one above and one below, on which were placed the weights for loading the bearings. The friction was measured either by the deflection of the compound pendulum thus formed, or as in most of the experiments by weighing its tendency to deflect by means of an attached cord running over a pulley and carrying a scale pan. See figure. The shafts or journals



Apparatus for Testing Roller Bearings.

used were of ordinary machinery steel, carefully turned to size and having a smooth finish. These shafts were rotated at the speeds shown by means of a belt and pulley. The cast-iron bearings used for comparison were cast whole and bored to size, but the babbitted ones were in halves and were held the same as the roller bearings.

In beginning an experiment a pointer on the lower end of the pendulum was brought to a zero mark vertically beneath the center of the shaft by adjusting the screws in the yoke. After the shaft began to revolve the pointer was held to the zero mark by putting weights on the scale pan. The product of the force thus applied to the pendulum by the distance of the point of application from the center of shaft gave the moment of friction, and dividing this by the radius of journal gave the friction at the surface of the journal. Dividing this again by the total weight on the journal gave the coefficient of friction.

In the first set of experiments Hyatt roller bearings were compared with plain cast-iron sleeves at a uniform speed of 480 revolutions per minute and under loads varying from 64 to 264 pounds. The cast-iron bearings were thoroughly and copiously oiled, the lubrication being rather better than would be the case in ordinary practice.

Table I shows the results of the test on one bearing in detail and from this it is seen that the value of f the coefficient of friction diminishes as the load increases, or in other words, the friction did not increase as fast as the load. This holds true as a general rule in all the roller bearings, but not generally in the plain bearings either cast iron or babbitt.

TABLE I.

Journal 1 15-16 inches in Diameter. 480 Revolutions per minute.

Total Load, pounds.	Friction.		Values of f.	
	Hyatt	Plain.	Hyatt.	Plain.
64.2	2.34	10.24	.036	.160
114.2	3.27	12.10	.029	.106
164.2	4.21	19.10	.026	.116
214.2	4.78	22.35	.022	.104
264.2	5.15	26.10	.019	.099
Average	....	....	.026	.117

Table II gives a summary of this series of experiments for the different sizes of journals, the different loads being the same as in Table I.

The relatively high values of f in the 2 3-16 and 2 15-16 roller bearings were due to the snugness of the fit between the journal and the bearing, and show the advisability of an easy fit as in ordinary bearings.

The same Hyatt bearings were used in the second set of experiments, but were compared with the McKeel solid roller bearings and with plain babbitted bearings freely oiled. The McKeel bearings contained rolls turned from solid steel and guided by spherical ends fitting recesses in cage rings at each end. The cage rings were joined to each other by steel rods parallel to the rolls.

The same apparatus was used as in the former tests, but heavier loads were used and the machine was run at a slightly higher speed.

TABLE II.

Values of Coefficient of Friction f. Speed 480 Revolutions per minute.

Diameter of Journal.	Hyatt Bearing.			Plain Bearing.		
	Max.	Min.	Ave.	Max.	Min.	Ave.
$1\frac{15}{16}$	.036	.019	.026	.160	.099	.117
$2\frac{3}{16}$	.052	.034	.040	.129	.071	.094
$2\frac{7}{16}$	.041	.025	.030	.143	.076	.104
$2\frac{15}{16}$	.053	.049	.051	.138	.091	.104

Table III shows the detailed results of experiments on one size of journal and is similar to Table I.

The last value given for the Hyatt bearing shows distortion of the roller due to the load and indicates the limit for this size. This is omitted in getting the averages.

There is the same indication as in Table I of a decrease of f with increase of load, and this was noticed in all the tests.

The results for the babbitt metal are not as uniform as the others on account of the difficulty of balancing.

TABLE III.

Journal 1 15-16 inches in Diameter. Speed of 560 Revolutions per minute.

Total Load.	Friction.			Value of f.		
	Hyatt.	McKeel.	Babbitt.	Hyatt.	McKeel.	Babbitt.
113.3	3.64	3.77	8.38	.032	.033	.074
162.3	3.77	4.24	8.97	.023	.026	.055
211.3	4.04	5.24	8.97	.019	.025	.042
260.3	4.31	5.37	8.97	.016	.021	.034
309.3	4.57	6.46	10.15	.015	.021	.033
358.3	4.71	6.73	10.75	.013	.019	.030
407.3	4.84	7.27	11.98	.012	.018	.029
456.3	37.70	7.81	20.90	....	.017	.046
Averages	....	....	....	.0186	.0225	.043

Under a load of 358.3 lbs. the solid rubber bearing showed an end thrust of about 20 lbs., which would account for the difference in friction between that and the Hyatt.

Table IV gives a summary of the tests in this series and may be compared with Table II.

The relatively high values for the Hyatt 2 7-16 bearing must be due to a slight clamping of the rolls due to too close a fit, as was noted in some of the former experiments.

Under a load of 470 lbs. the Hyatt bearing developed an end thrust of 13.5 lbs. and the McKeel one of 11 lbs. This end thrust is due to a slight skewing of the rolls and would vary, sometimes even reversing in direction.

The babbitt bearing is a slight improvement over the cast-iron sleeve, but the difference is quite as apt to be due to improved lubrication (notice the variation in the averages for the various sizes in Table IV).

TABLE IV.

Values of Coefficient of Friction f. Speed 560 Revolutions per minute.

Diameter of Journal.	Hyatt Bearing.			McKeel Bearing.			Babbitt Bearing.		
	Max.	Min.	Ave.	Max.	Min	Ave.	Max.	Min.	Ave.
$1\frac{15}{16}$	.032	.012	.018	.033	.017	.022	.074	.029	.043
$2\frac{3}{16}$	.019	.011	.014	......	......	......	.088	.078	.082
$2\frac{7}{16}$	.042	.025	.032	.028	.015	.021	.114	.083	.096
$2\frac{15}{16}$	.029	.022	.025	.039	.019	.027	.125	.089	.107

In conclusion it may be said that the friction of the roller bearing is shown to be from one-fifth to one-third that of a plain bearing at moderate loads and speeds. It is also noticeable that as the load on a roller bearing increases the coefficient of friction decreases.

It was found by the experimenters that a slight change in the pressure due to the adjusting nuts was sufficient to increase the friction considerably. In the McKeel bearing the rolls bore on a cast-iron sleeve and in the Hyatt on a soft steel one. If roller bearings are properly adjusted and not overloaded a saving of from 2-3 to 3-4 of the friction may be reasonably expected.

* * *

Once there was a foreman who believed that "economy is wealth," and so thoroughly did he believe this maxim that he required his toolmakers to wear their files down to the last degree. Cochrane was making an unusually large die and the foreman's policy caused him to spend two or three weary days in filing out the die that would have been unnecessary if good, sharp files had been provided. He then and there resolved that when he escaped from the drudgery of the job he would devise a filing machine that would save other unfortunates from such slavery. The result is a machine which does the work of the human operator with an accuracy that the latter could never hope to equal. It is true that it works more efficiently with sharp files than dull ones, but if it is forced to use dull ones its cheerfulness is not at all affected. In this machine we have another proof, though none is needed, that it is the average American's unconquerable aversion to unnecessary manual labor that causes the activity of his "think tank" and the invention of the thousand and one devices calculated to lighten labor and to increase its efficiency.

* * *

When a crank-pin is forced into its disk it is held in place by the tension of the disk compressing the particles of the crank-pin, with such a grip that the friction usually holds it firmly against any ordinary strain. In fact, a crank-pin is usually held *too* firmly when for any reason it becomes necessary to remove one in the engine room, for the appliances necessary for generating the required pressure are rarely at hand. It does not avail to heat the parts for the pin and disk expand at practically the same rate so that the tension is not changed; but what may be done in case a pin is useless for further service, is to drill a hole through it longitudinally and tap each end of the hole for pipe connections. Then when the disk and pin are heated, the pin may be cooled rapidly from the interior and thus shrunk so that it may be pulled out with comparative ease. One of the contributors to a paper called "Wrinkles," which was presented at the June convention of the National Electric Light Association, describes this plan which was successfully employed by him.

THE WORM GEAR.

JOHN EDGAR.

The worm gear has been bothering machine designers ever since its conception, having been discarded as unsatisfactory time and again, only to come to the front in further attempts to make itself useful. The last period of discussion was in relation to the helix angle. Mr. Halsey in his articles in the *American Machinist* on this subject has given data from a variety of different successful and unsuccessful examples, which show that the helix angle gives the highest efficiency when above 12 degrees, increasing gradually up to 45 degrees, and that all examples with an angle less than 9 degrees proved unsuccessful.

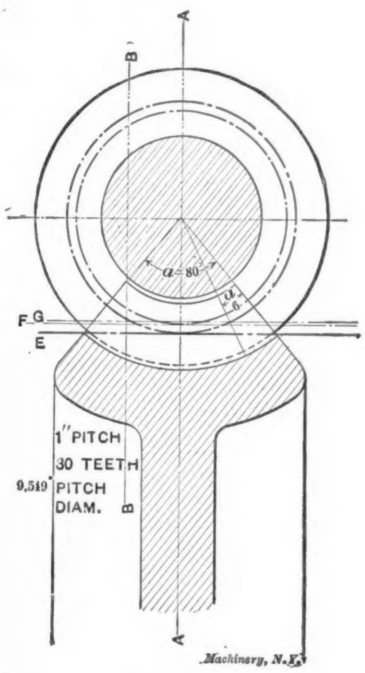


Fig. 1.

In the course of his articles no mention was made of the sizing of the gear blanks, so that one may rightly come to the conclusion that they were sized according to standard practice. This method locates the pitch line of the worm on a circle whose radius is smaller than that of the worm by an amount equal to one-half the working depth of the tooth. Where the working depth, as in standard practice, is equal to .6366 times the linear pitch, and when P' is the

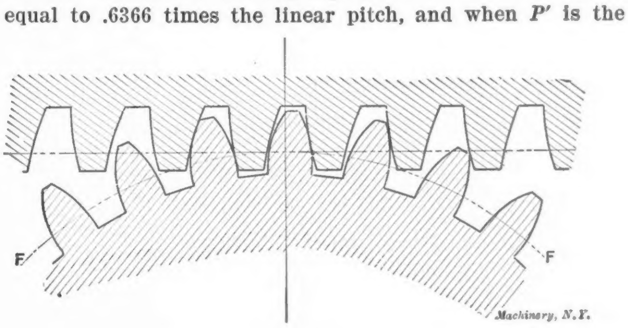


Fig. 2.

linear pitch, D the outside diameter and D' the pitch diameter of the worm, this fact may be expressed by the following formula:

$$D' = D - .6366\,P'. \qquad (1)$$

In Fig. 1 we have a section through a worm and worm gear. The pitch circle for the worm according to standard practice is located as shown tangent to the line E, which is the pitch line of the worm gear. On inspection of the figure it is seen

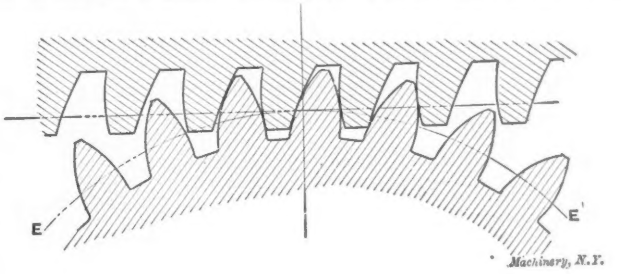


Fig. 3.

that while the addendum of the worm and worm gear are equal, at the center line $A\,A$, they are not at any other point along the pitch line, either to the right or the left: A section taken through the gear on the line $A\,A$ would reveal teeth similar in shape to those of a spur gear of the same pitch and number of teeth. But how does this shape vary as we shift

this line either side of the central position? Let us show this by example, taking the case of a worm having a single thread of 1-inch pitch. By taking a section on line *B B* instead of the center line *A A* we obtain Fig. 2. This figure shows plainly that the faces of the teeth of the gear are considerably longer than the flanks. It is easily seen that the greater the angle α is, the greater will this difference be, and *vice versa*, until we reach the central position, where there is no difference. Therefore we see that this angle α plays an important part in the design of a successful worm gear.

This angle is not the only cause of distortion in the shape of the tooth. With a little thought it will be seen that the angle of the helix also is cause for further irregularity. To illustrate this we will take the case of a worm having the same pitch, but having three threads instead of one, giving a lead of 3 inches. A section of this at *B B* is shown in Fig. 3. These conditions have the effect of producing even longer faces than do those in the former case.

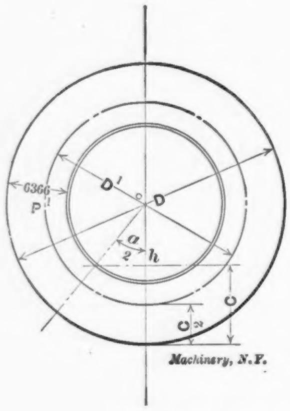


Fig. 4.

What can be done to remedy this defect? We can shorten the faces, but when we do that at this point we do so all along the face of the gear and thus change the shape at *A A*, where it is normal. Therefore, the best we can do is to divide the difference at the two extreme points—*A A* and *B B*. This can be done as follows: In an ordinary spur gear of standard proportions the pitch line is located at a point midway of the working depth. From Fig. 4, which shows the end view of a worm, we see that the total working depth is equal to *C*, so that from the foregoing statement the pitch line should pass through a point situated at a distance equal to one-half of *C* from the outside of the worm making D' the pitch diameter of the worm.

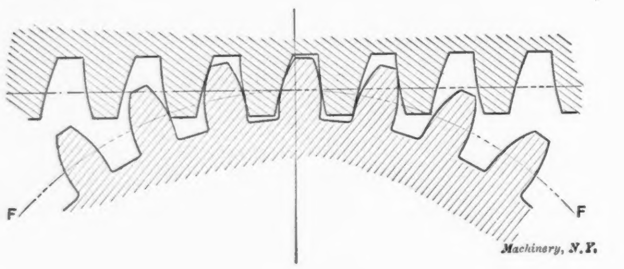


Fig. 5.

By an inspection of Fig. 4 we may derive the following formula:

$$C = \frac{D}{2} - \cos\frac{\alpha}{2}\left(\frac{D}{2} - .6366\,P'\right) \qquad (2)$$

Since $D' = D - C$, we may obtain the value of D' in terms of D, P' and α:

$$D' = \frac{D}{2} + \cos\frac{\alpha}{2}\left(\frac{D}{2} - .6366\,P'\right) \qquad (3)$$

Solving this last equation for D, we have the means for finding the outside diameter when D', P' and α are given:

$$D = \frac{2\,D' + 1.273\,P' \cos.\dfrac{\alpha}{2}}{1 + \cos.\dfrac{\alpha}{2}} \qquad (4)$$

Formulas 3 and 4 may be used for obtaining the pitch diameter of any worm when the outside diameter is known, and *vice versa.*

It is quite evident that the method given by Mr. Perrigo in the June issue for obtaining the pitch diameter of the gear is based on this principle, but it is only an approximation, the variance between its results and those of the formula increasing with the angle α. The difference for the example we have been investigating will be seen in Fig. 1 where *G* is the line as located by his method, *F* that by the formula; and *E* the standard location.

To show the difference this change in location of the pitch line makes in the tooth shape as compared with the usual practice, I have drawn sections at *B B* for both a single and a triple threaded worm of 1-inch pitch. Figs. 5 and 6, respectively, show these sections. Here we see that while the faces are yet considerably longer than the flanks the shape is improved. The difference between Fig. 5 and a normal section is very slight and hardly noticeable, and while the shape in Fig. 6 is somewhat freakish it has all the properties of a smoothly running gears.

But someone may ask what all this has to do with the durability of the gear. It is this: It has been proved that the friction of approach is much more in amount than that of the release. This friction of approach occurs between the face of the driven gear and the flank of the driver. Now if these particular elements of the tooth are extra long, the friction is proportionately increased over what it would be in a normal tooth. The friction of motion is always accompanied by wearing of the surfaces in contact; therefore in order to increase the life of the gear we must decrease the friction to a minimum. This we have done by locating the pitch line in accordance with the formula.

In order to illustrate the extent to which some designers go to eliminate the friction between the surfaces of the teeth in contact, I might cite the case of some special forms of clock gearing, where the driver is made with teeth having no flanks and the driven gear with teeth having no faces, fixing all the contact at the period of release. The reader may prove the importance of this point for himself by observing the wear on the teeth of a pair of gears that run constantly in one direction.

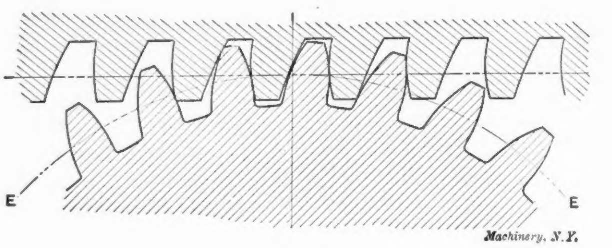


Fig. 6.

It would be interesting if Mr. Perrigo, and any others who have at hand data on successful and unsuccessful examples of this form of gearing, would analyze some of the most prominent cases. It would prove instructive as well as interesting.

The tooth curves in the above figures were obtained by the tracing cloth method described in Unwin's Machine Design, which time and space hinder me from giving here.

I would suggest that all those who have not done so, should obtain a copy of *Mr. Halsey's articles on the worm gear, as he gives some very important data, showing how unsatisfactory examples have been altered so as to prove satisfactory. All the examples have been taken from practice and have been in use in all parts of the country.

* * *

The handling of a railway car as though it were a huge spoon is an accomplishment which is daily demonstrating its effectiveness at coal wharves and other places where a large number of gondola coal cars have to be unloaded. The cars are progressively run into an unloading platform, which, after the car is rigidly clamped fast, is lifted and tilted to one side until the contents all pour out. A 50-ton car by this process can be clamped, lifted and tilted and returned to its original place in less than two minutes. To a certain extent the converse of this action is successfully being employed for loading box cars with coal, but the cradle, instead of tilting sideways, tilts longitudinally; thus allowing the coal which runs from a chute into the side door, to run first to one side of the car and then to the other as the car is tilted in the opposite direction.

* "Worm and Spiral Gearing," Frederick A. Halsey. Published by the D. Van Nostrand Co., New York. Price 50 cents.

CUTTING SPIRAL GEARS.

E. H. FISH.

E. H. Fish.

In taking up the subject of spiral gears with students at the Worcester Polytechnic Institute, we have experienced some difficulty over the formulas relating to their construction. If these trouble men accustomed to the use of trigonometry, they must certainly be confusing to shop men. As the explanation of the method of figuring these gears, which we have arrived at for the student's use, involves a minimum amount of mathematics, we feel that it may be of value to others.

As it is more convenient to adapt these gears to the standard diametral pitch cutters used for spur gears, we will consider the subject only from that point of view. This gives us at once the normal pitch; that is, the pitch measured perpendicular to the face of the tooth, and also the shape and depth of the tooth.

In the case of a spur gear, we cut the teeth at right angles with the base of the cylinder on which the gear is cut, as at a in Fig. 1. The space appears in its true size and shape on the base of the blank. If, now, we cut the teeth at some other angle, say at 30 degrees with a line parallel to the axis of the gear blank, as at b in Fig. 1, we see that the width of the space measured on the base is greater, and it will be greater still if the angle is increased. It is thus evident that the number of teeth that can be cut on a given cylinder decreases as the angle of the teeth with a line parallel with the axis of the gear increases.

Number of Teeth and Diameter of Blank.

Referring to Fig. 2, suppose the line $b\ c$ to be a part of the base of the cylinder and the two lines making the angle

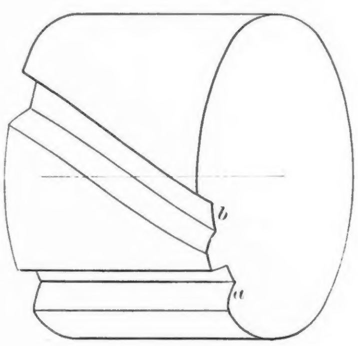
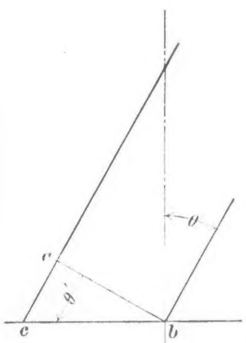


Fig. 1.



Machinery, N.Y.

Fig. 2.

θ ($=\theta'$) with a line parallel with the axis to represent the center lines of two adjacent teeth. Then, $a\ b$ will represent the normal pitch, and $b\ c$ the circumferential pitch; but

$$a\,b = b\,c\ \cos\,\theta,\ \text{whence}$$

$$b\,c = \frac{a\,b}{\cos\theta}.$$

The number of spur-gear teeth that can be cut in a blank of pitch radius, r, is expressed by the formula:

$$N = 2\,r\,P, \qquad (1)$$

where P is the diametral pitch and N the number of teeth.

From this we see that the number of teeth in a spiral gear of this pitch and pitch radius, and of angle θ, will be

$$N' = 2\,r\,P\cos\theta. \qquad (2)$$

Take as an example a gear to be cut 6-pitch, with teeth at an angle of 60 degrees to a line parallel with the axis of the gear; pitch diameter to be about $2\frac{1}{2}$ inches. Then, $r = 1\frac{1}{4}$; $P = 6$; $\cos\theta = 0.5$. Hence,

$$N' = 2 \times 1\tfrac{1}{4} \times 6 \times 0.5 = 7\tfrac{1}{2}.$$

As a gear of $7\frac{1}{2}$ teeth is impracticable for continuous rotation, we must make the number of teeth either 7 or 8. Suppose we make it 8. Then, to find the pitch diameter of the gear we use the same formula, but transposed as follows:

$$r = \frac{N'}{2\,P\cos\theta},$$

from which we get, after substituting 8 for N',

$$r = \frac{8}{2 \times 6 \times \frac{1}{2}} = 1\ 1/3.$$

The pitch diameter of our blank must, therefore, be 2 2-3 inches, the same as for a spur gear of 16 teeth. As we are using diametral pitch cutters, the addendum will be the same as for a spur gear of the same pitch. Adding $1/P$ to the pitch diameter on each side will make the whole diameter, 3 inches in this case.

Milling Spiral Teeth.

In order to mill the teeth, we must be able to set up the machine so to make, approximately, the correct advance per revolution of the work. This advance will be equal to the circumference of the blank, measured on the pitch line multiplied by the cotangent of the angle of the teeth. As this usually presents no difficulty to student or workman, we pass it over with simply saying that gears run together quite nicely, even if the lead as figured is not exactly obtainable on the milling machine. A difference of 4 or 5 per cent is not very noticeable.

After having set our machine to cut the desired spiral, we next wish to select the proper cutter. This will be, unless the angle θ is very small, quite a different cutter from that used for a spur gear of the same diameter, or of the same number of teeth. Brown & Sharpe advise turning up a blank of the size of the pitch diameter and laying on it a helix at right angles to the helix of the teeth of the gear to be cut, as in Fig. 3, fitting a cardboard template to the face of the cylinder along this curve, and then finding the diameter of the circle corresponding to this template.

The cutter should be such as will be suitable for a gear of this diameter and the given normal pitch. This is a sufficiently close method for gears of a large number of teeth, but requires considerable care for gears of 12 or less teeth. Moreover, we require a method that can be worked out entirely in the drafting room. Grant says that the cutter should be right for a spur gear having a number of teeth equal to the number of teeth in the spiral gear, divided by the cube of the cosine of the angle of the teeth. This gives an exact result, but he offers no explanation of his statement. The following, we hope, will seem a clear demonstration:

Demonstration of Grant's Formula.

It will be seen that what we wish to find at the start is a circle having the same radius as the helix, which is drawn on our pitch cylinder perpendicular to the teeth, as in Fig. 3. The angle of this helix will be $90 - \theta$ degrees. If $R =$ radius of curvature of this helix, then from the well-known formula of analytic geometry for the radius of curvature of a helix, we have

$$R = \frac{r}{\sin^2(90 - \theta)} = \frac{r}{\cos^2\theta}. \qquad (3)$$

The demonstration of this formula will shortly be given for the benefit of those who enjoy mathematics.

Referring, now, to formula (1) and applying it to a gear of radius R, we have

$$N = 2\,R \times P = \frac{2\,r}{\cos^2\theta} \times P. \qquad (4)$$

For our spiral gear we found, by formula (2), that:

$$N' = 2\,r\,P\cos\theta.$$

Dividing (4) by (2), we have

$$\frac{N}{N'} = \frac{2\,r\,P}{\cos^2\theta} \times \frac{1}{2\,r\,P\cos\theta} = \frac{1}{\cos^3\theta},$$

E. H. FISH was born in Worcester, Mass., 1870. He graduated from the Worcester Polytechnic Institute in 1892, after which he worked for the H. C. Fish Machine Works. He has held positions as foreman, draftsman, superintendent, and for the past two years instructor of drawing and machine design at Worcester Polytechnic Institute. His specialty is the design of machine tools.

or, $$N = \frac{N'}{\cos^3 \theta}.$$

Since N' is the number of teeth in our spiral gear and N is the number of teeth in a spur gear which has the same radius as the radius of curvature of the helix above referred to, this is the equivalent of saying that the cutter to be used should be correct for a number of teeth which can be obtained by dividing the actual number of teeth in the gear by the cube of the cosine of the tooth angle. Since the cosine of angle θ is always less than unity, its cube will be still less, so N is certain to be greater than N', which will account for the fact that spiral gears of less than 12 teeth can be cut with the standard cutters. The getting of the cube of $\cos \theta$ may bother some, as the cubing of any fraction is apt to do, but a graphical method is given later in the article which, even if roughly laid out, will give sufficiently accurate results for this purpose. For the other uses of this graphical method, care must be used, or the results are not to be depended on.

Calculation of Velocity Ratio.

So far, we are able to cut the gear, once having decided on the number of teeth, pitch (or pitch diameter) and angle of

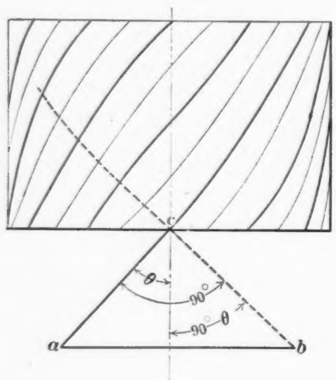


Fig. 3.

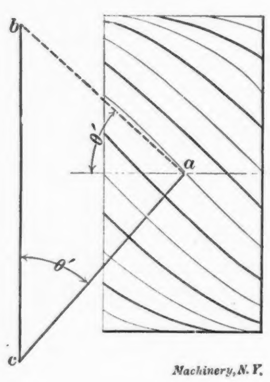


Fig. 4.

teeth, but in designing we almost always wish to transmit motion with some definite velocity ratio. If we were dealing with spur gears we would know that the ratio of speeds would be inversely proportional to the pitch diameters or the number of teeth. If the teeth were twisted or cut spiral on the surface and the axes were left still parallel, this same velocity ratio would obtain, but the moment we move the axes out of the same plane this convenient ratio ceases to exist. Then there can be but one point of contact of the pitch cylinders, consequently all motion must be transmitted as if through this one point if smooth running is to be attained. The actual motion of the tooth at this point must be at right angles to the axis of the gear, but it may be considered as the resultant of two motions, one of sliding parallel to the teeth, which we can see must happen since the two gears do not run in the same plane, and the other perpendicular to the teeth, which is the effective, or driving motion. This latter motion normal to the teeth must be same for both gears.

In the case of a driving gear of radius r and angle θ, the velocity of this point in a plane perpendicular to the axis will be $2 \pi r n$, where n is the number of revolutions per minute of the driving gear. Let us consider the point c of the gear in Fig. 3. Assume the line $a\,b$ to represent the linear velocity and to be equal to $2 \pi r n$. The line $c\,b$ is perpendicular to the tooth, and $a\,c$ parallel to the tooth. These three lines complete the triangle $a\,b\,c$, and therefore $a\,c$ will represent the sliding component of the point c, and $c\,b$ the motion perpendicular to the tooth. Then,

$$b\,c = 2 \pi r n \cos \theta,$$

since in the triangle $a\,b\,c$ the angle $a\,b\,c$ is equal to θ.

This, also, is the velocity of the contact point of the driven gear in the same direction, or in a direction normal to the teeth of the driven gear. We will assume this gear to have a radius r' and angle θ'. Considering the gear in Fig. 4 to be the driven gear, with axis at right angles to the axis of the driving gear, we have

$$a\,c \text{ (Fig. 4)} = b\,c \text{ (Fig. 3)} = 2 \pi r n \cos \theta.$$

The resulting motion perpendicular to the axis of the gear will then be

$$c\,b \text{ (Fig. 4)} = \frac{a\,c}{\cos \theta'},$$

$$= \frac{2 \pi r n \cos \theta}{\cos \theta'}$$

This is the linear velocity of the point a; to get the number of revolutions of the driven gear we divide by the circumference of the driven gear, which is $2 \pi r'$, giving

$$n' = \frac{2 \pi r n \cos \theta}{2 \pi r' \cos \theta'},$$

$$\text{whence } \frac{n'}{n} = \frac{r \cos \theta}{r' \cos \theta'}.$$

That is, the relative motion of the two gears is inversely proportional to the product of their diameters and the cosines of the angles of their teeth.

If both are 45-degree gears, this last factor becomes inoperative, and the gears produce motion in the same ratio as spur gears of the same sizes. The same is also true if the axes are parallel, for θ and θ' then become equal.

If the axes are at right angles, $\theta - 90 = \theta'$, and $\dfrac{\cos \theta}{\cos \theta'} = \dfrac{\cos \theta}{\sin \theta} = \cot \theta$, whence: $\dfrac{N}{N'} = \dfrac{r}{r'} \times \cot \theta$.

This property of spiral gears, of having a varying velocity ratio for both size and angle, is valuable, in that it enables one to obtain varying velocity ratios with the same size gear. For example, suppose we have two gears, one of 8 teeth and one of 16 teeth, both 45-degree gears, on axes at right angles. The velocity ratio is 2 to 1. If, now, we want a velocity ratio of 3 to 1 on the same axes with the same size gears, we use the formula last arrived at,

$$\frac{N}{N'} = \frac{r}{r'} \cot \theta, \quad \text{or,} \quad \frac{1}{3} = \frac{1}{2} \cot \theta,$$

$$\cot \theta = \frac{2}{3} = 0.6666.$$

θ will then be 56° 19′ and θ' will be 33° 41′.

If we use cutters of the same pitch as before, N and N' will become fractional numbers, thus making impossible condi-

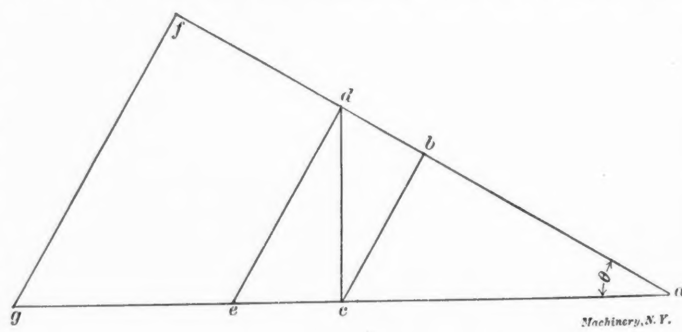


Fig. 5.

tions for practical use. It will, then, be necessary to use a fractional pitch cutter. To find what this cutter should be, decide on the number of teeth to be used in each of the two gears to give the desired new velocity ratio of 3 to 1; then solve formula No. 2 for P, substituting the required data from either of the two gears.

The relation between the angles of the shafts and gear teeth will be readily understood by a little thought. In gears whose axes are at right angles we have seen that the sum of the angles of the gear teeth is equal to 90 degrees, the angle of the shafts. This is true for any gears whose spirals are both right-hand or both left-hand. Carrying this to an extreme, we find that if the tooth angles become zero degrees (as in spur gears), the shaft angle becomes 180 degrees, or the shafts are parallel. If one gear is right-hand and the other left-hand, then the angle of the shafts will be equal to the difference of the tooth angles. If the gears have their teeth at equal angles, but one right-hand and one left-hand, then the shaft angle will

be zero; that is, the shafts are parallel and the gears are twisted gears, or Hooke's gears.

The annexed figure, while it is innocent looking enough, contains a solution of all the bothersome points of the figuring of spiral gears to be cut with the usual diametral pitch cutters.

To illustrate the use of the figure, we will take as an illustration a 24-tooth gear of 30-degree spiral angle, to be cut with an 8-pitch spur-gear cutter.

Lay off $a\,b = 3$ inches, the diameter of a spur gear of 24 teeth, 8-pitch. Lay off the angle θ 30 degrees as shown and erect a perpendicular at b to $a\,b$, intersecting $a\,c$ at c. The line $a\,c$ will be the pitch diameter of the required spiral gear (3.46 inches). The outside diameter will be equal to this diameter plus $\frac{2}{P}$, as in spur gears (3.71 inches). The depth of tooth will be the same as for a spur gear of the same pitch.

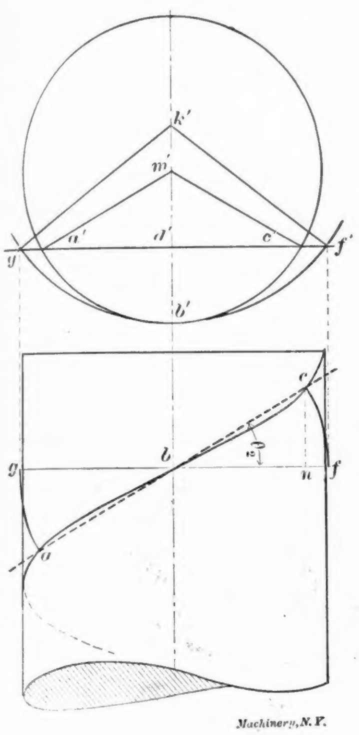


Fig. 6.

Extend $a\,b$ and $a\,c$. At c erect a perpendicular to $a\,c$, meeting $a\,b$ at d. At d, in turn, erect a perpendicular to $a\,d$, meeting $a\,c$ at e. $a\,e$ will be the diameter of a spur gear having the correct number of teeth from which to choose a cutter to cut our spiral. In this case the diameter is 4⅝ inches, corresponding to a 37-tooth gear. So we will use the same cutter to cut our 24-tooth spiral gear as that we would use to cut a 37-tooth spur gear.

Extend, in turn, $a\,d$ and $a\,e$ till a line, of length equal to $a\,c$, drawn perpendicular to $a\,d$ will just meet $a\,e$, as $f\,g$; then $a\,f \times \pi$ will be the pitch of the spiral to which we should set the milling machine (in this case, 18.85 inches). The diagram depends on the following facts relating to spiral gears.

(1) $$\frac{b\,a}{a\,c} = \cos\theta = \frac{\text{diam. spur gear}}{\text{diam. spiral gear}}.$$

(2) $$a\,c = \frac{b\,a}{\cos\theta}$$

$$a\,d = \frac{a\,c}{\cos\theta} = \frac{b\,a}{\cos^2\theta}$$

$$e\,a = \frac{a\,d}{\cos\theta} = \frac{b\,a}{\cos^3\theta}.$$

which corresponds to equation 3 above.

(3) $$\frac{\text{Pitch of first helix}}{\text{Circumference of pitch cylinder}} = \tan\,(90^\circ - \theta)$$

Divide by π and transpose

$$\frac{P}{\pi} = \text{diameter of pitch cylinder} \times \frac{f\,a}{f\,g}$$

$$P = \frac{f\,a}{f\,g} \times (a\,c = f\,g) \times \pi = f\,a \times \pi$$

Therefore pitch of first helix $= f\,a \times \pi$.

Proof of Formula No. 3.

For the proof of formula No. 3, $R = \frac{r}{\cos^2\theta}$, where R = radius of curvature of a helix.

r = radius of the cylinder on which the helix is drawn, and θ is the angle of the helix with a line parallel to the axis of the cylinder, we have:

In Fig. 6 is a cylinder of radius $m'\,c' = r$, on which is drawn a helix. We have assumed three points, a, b and c equidistant on the helix, the middle point, b, being taken at the extreme front of the helix, for convenience only.

We wish to draw a circle passing through the three points, a, b and c. To do this we have revolved the two outside points into the same horizontal plane as b, placing a at g and c at f. We represent these points in the top view by g' and f'. Through g', b' and f' we draw a circle having its center at k' and radius $k'\,f'$, which we will call R_2. This circle will be represented in the front view by the horizontal line g to f. The original position of this circle in the front view is represented by the straight line a to c. The angle of these two lines we will call θ_2. Remember that this is not the angle of the helix with the base, but is the angle of the original plane of the circle through a, b and c with the horizontal.

(1) $b\,n = d'\,c' = b\,c\cos\theta_2$

(2) $b\,c = b\,f = d'\,f'$

Then,

(3) $d'\,c' = d'\,f'\cos\theta_2$

Squaring,

(4) $(d'\,c')^2 = (d'\,f')^2\cos^2\theta_2$

(5) $(d'\,c')^2 = (m'\,c')^2 - (m'\,d')^2 = r^2 - (m'\,d')^2$

(6) $(d'\,f')^2 = (k'\,f')^2 - (k'\,d')^2 = R^2_2 - (k'\,d')^2$

Substituting (5) and (6) in (4), we have,

(7) $r^2 - (m'\,d')^2 = [R^2_2 - (k'\,d')^2]\cos^2\theta_2$

$m'\,d' = r - d'\,b'$

(8) $(m'\,d')^2 = r^2 - 2\,r\,(d'\,b') + (d'\,b')^2$

$k'\,d' = R_2 - d'\,b'$

(9) $(k'\,d')^2 = R_2^2 - 2\,R_2\,(d'\,b') + (d'\,b')^2$

Substituting from 8 and 9 in 7 we get:

(10) $r^2 - r^2 + 2\,r\,(d'\,b') - (d'\,b')^2 =$

$$[R_2^2 - R_2^2 + 2\,R_2(d'\,b') - (d'\,b')^2]\cos^2\theta_2$$

cancelling we have,

(11) $2\,r - d'\,b' = (2\,R_2 - d'\,b')\,\text{cor}^2\,\theta_2$.

This expression is true for any three points equidistant on the helix. Let us remember that the radius of curvature for any curve is the radius of the circle passing through any three consecutive points. We will accordingly consider points a and c moved up so that they become consecutive points with b and see what the effect is on equation 11.

r will remain constant.

$d'b'$ will become practically zero on each side of the equation and may be neglected.

R_2 becomes R, the radius of curvature of the helix, and

θ_2 becomes θ, the angle of the helix.

Substituting these values in (11), we have,

(12) $2\,r = 2\,R\cos^2\theta$

$$\text{or, } R = \frac{r}{\cos^2\theta}$$

* * *

4,000 KW. GAS ENGINE DRIVEN GENERATOR.

The station of the California Gas and Electric Corporation used for the operation of the United Railways of San Francisco will shortly be equipped with three generators of 4,000 kilowatts capacity each, driven by gas engines. The latter, which will be the largest yet made, will be built by the Snow Steam Pump Co. The generators will be built by the Crocker-Wheeler Co., and will be of the type developed by Brown, Boveri & Co. in Switzerland. The generators will be capable of working as motors in starting the engines that drive them.

* * *

One of the surprising things learned about friction in the investigations conducted with the air-brake is that the coefficent of friction continuously and regularly declines as the speed increases. The observed coefficient of friction between a cast-iron brake shoe and a chilled wheel is 0.111 at 55 miles per hour, and from the data acquired it is calculated that the coefficient would be 0.105 for 60 miles per hour; 0.085 for 85 miles per hour; and 0.072 for 100 miles per hour. In short, the frictional resistance of a bearing should be less with no lubricant at all when running with a peripheral speed of 100 miles per hour than it is usually with the best lubricants, running at ordinary speed.

THE NEW GISHOLT FOUNDRY.

About a year ago the Gisholt Co., Madison, Wis., completed a new foundry building, which is a good example of a modern foundry equipment, designed for the needs of a good-sized machine tool business. The main building is 120 × 240 feet, and is constructed with a steel frame and brick skin, therefore being of fireproof construction. No wood whatever was used in the building, save in the window frames and doors. The building is located on a plot of such size that future additions may be made in harmony with the general plan of plant enlargement. At present only one cupola has been provided, which has a capacity of 20 tons per hour. Space has been provided for a second cupola when the need shall demand it. The floor is served by a 20-ton electric traveling crane, and electric hoists, sand shakers, special iron flasks, core-making machines, etc., are a part of the up-to-date equipment. The exterior view, Fig. 1, shows that a large part of the wall space is glass, and the lighting is consequently very good. A feature of the roof construction is that the glass sides of the monitor are set in an inclined position, thus probably giving a considerably better diffusion of the light coming from this source than is possible when the sides are vertical. Nearly one-third of the top of the monitor roof is glass which, with the other sources, causes the light to come in from all directions, thus avoiding the casting of dense shadows on almost any part of the foundry floor; in short, a man does not stand in his own light, no matter in what part of the building he may be working, or scarcely in what position. This condition is evident from the view, Fig. 2, which should convey some impression of the extreme contrast between this building and the typical foundry interior.

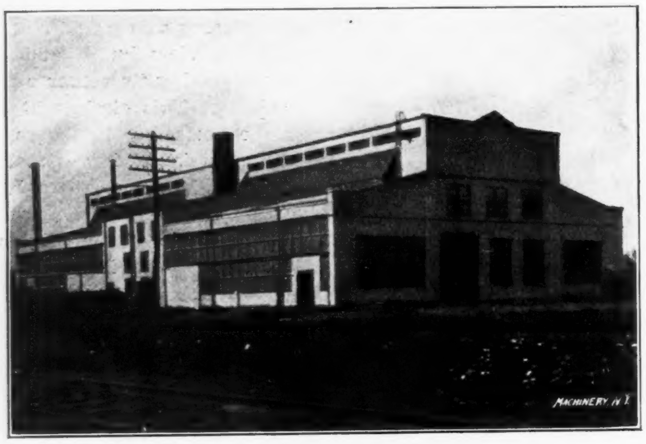


Fig. 1. The New Foundry of the Gisholt Co.

Adjoining the foundry, and connected with it by a narrow gage track, is the casting storage building, a structure 100 feet wide and 200 feet long. The storage building is equipped with a traveling crane and hoists for handling heavy castings. The tumbling and cleaning rooms are in a separate building and the tumbling barrels are electrically driven. The pattern shop is in another building, 125 feet wide and 210 feet long. The system of buildings is connected by an industrial railway which connects it with the main machine shop across the street. Standard gage switches connecting the foundry and machine shop are also provided from the adjoining tracks of the Chicago, Milwaukee & St. Paul Railway.

* * *

The matter of taking up foreign patents is often a perplexing one to a manufacturer, as in many cases he does not expect to exploit the foreign market, and even if he does so, he is willing to do it in the open market without the advantage of foreign patent protection. The cost of taking out foreign patents, and the taxes and other charges, in some countries make them a burden to such an extent that it is often considered unprofitable to take them out, even on meritorious inventions. In a recent report, Mr. Frank Mason, United States Consul-General at Berlin, Germany, refers to the foreign agents' side of the question. From this point of view, it is very desirable that American goods which are to be sold in foreign markets, especially in Germany, shall be patented in those countries. It is hardly fair to ask the selling agent to spend his time doing missionary work in securing a market for unpatented articles, for as soon as it has become introduced, other manufacturers will at once enter the field, and rob the selling agent of the fruits of his endeavor. Then, again, there is the customer to consider, who runs the risk of buying a lawsuit with his machine or other purchase.

Fig. 2. Interior of the Foundry.

TECHNICAL READING AND FORMULAS.—1.

C. F. BLAKE.

I. General Remarks on Self-Education.

C. F. Blake.

There are several ways of obtaining an education: The easiest and, until recent years, the usual way is to begin at the age of seven and continue steadily at school till the age of twenty-four, at father's expense. It is a fortunate fact that education is by no means unattainable otherwise; indeed many of the greatest and most widely useful educations the world has known have been obtained almost without a look at the inside of a school. A second method, quite modern, is the correspondence school—most excellent in many respects, yet not completing the available ways of obtaining an education. For the purposes of this paper the final method is that of self-education. The writer has obtained a very large share of his education in this manner, and has, as well, assisted others to do the same, and it is to explain the possibilities of this method and to plant the seed of self-help elsewhere that these papers are undertaken. They are divided into four heads dealing with the following subjects:

1. Present introduction, explaining general methods to be followed.
2. Technical reading, journals, catalogues and formulas.
3. Strains set up in machine materials, and strength of materials to resist them.
4. Graphical methods in use in machine designing and technical writing.

This is an alluring subject, and it is difficult to know where to draw the line. Indeed it would be impossible to stop were it not for the limitations of space, as well as the deep-seated idea the writer holds that nothing is to be considered worth while that is made too easy. Consequently it is not our intention, even were it possible, to open wide the door of knowledge. It is the aim of these papers to start the ambitious young man of sufficient grit upon a path which, if rightly followed, will in the future surely place him on par with those more fortunate men of his age who have enjoyed a college education, and to leave him in a position to continue to read and study and to understand the technical discussion and articles on design which appear in the technical press.

Engineering education does not consist in knowing things mechanical—very far from it. It consists only in knowing where to find technical literature upon any given subject when it is wanted, and knowing how to read it when it is found. Therefore the first thing needed by our student is a place to store his newly acquired knowledge, aside from his head. The writer's first attempt in this line resulted in a book having black canvas covers and a flexible back. Tapes were provided to lace in the leaves, which were made of fairly heavy cardboard, perforated for the tapes, and having a flexible strip along the perforated edge to enable the leaves to turn back properly.

Twenty-six alphabet leaves were made similar to those in dictionaries and memorandum books, and a supply of extra leaves kept on hand.

Clippings from papers and catalogues were pasted on blank leaves and inserted under the proper letter, also notes and formulas received from others were written in, making the book a record of past work and study. The book, finally becoming too large to be convenient and too small to hold everything to be preserved, gave way to the card index and filing case. A card index outfit large enough to answer all requirements for a long time may be purchased for $1.25 and a filing box for $1.50. These include all index cards, blank cards, and envelopes for clippings.

Having provided a systematic way to file our clippings, we are ready to consider the sources of the same. First subscribe for one or two of the leading technical journals devoted to your line of work. Make a practice of sending for catalogues of machinery manufacturers, and file them in the filing box. Many catalogues present, besides the goods manufactured, tables and data of value. If you can clip out these tables and file them in the card index without destroying the catalogue, do so; if not, make an entry in the card index to show where they may be found before filing the catalogue. Always write your name in the catalogues, for as the file grows, you will find demands upon it from others, and this will aid in keeping the file intact.

Remember that a catalogue received implies confidence on the part of the sender that it will eventually prove of use to him by bringing his goods before possible purchasers, and for this reason, as well as for your own convenience, all catalogues received should be listed and filed.

Duplicate clippings, such as tables, may often be exchanged with others, and thus our files are enlarged. This is not meant to encourage a mere mania for collecting—far from it. We should so study all data filed as to understand it at the time, and if found difficult, make such notes as will readily recall the study to our minds in the future.

II. Technical Reading and Formulas.

The first thing to be done in preparation for study, and for reading the technical papers, is to become familiar with the *engineering language*. The *spoken engineering language* is of course the native tongue of the country, with, however, plenty of new words to master; but the *written engineering language* consists very largely of symbols, so like those of higher mathematics in appearance as often to discourage the beginner from further efforts. In the *written engineering language* rules, instead of being written in the native tongue, are expressed by combinations of these symbols, and when so expressed are called formulas.

Now, the mathematician when deriving a formula, uses the same symbols as the engineer when writing a formula, and if we accept the work of the mathematician as correct, we need pay no attention to the use of these symbols in deriving formulas, but give our attention to learning to read the symbolic language of the engineer with sufficient ease to enable us to follow the operations called for by any formula we may wish to use.

The following table exhibits in the first column the symbols most frequently met with; in the second column the arithmetical equivalent of the symbols is given, assuming that $a=2$ and $b=4$; in the third column the symbols are expressed in English to give the proper method of reading the symbols.

TABLE NO. 1.

$a = 2 \quad b = 4$		a equals 2 b equals 4
$a + b = c$	$2 + 4 = 6$	a plus b equals c
$b - a = d$	$4 - 2 = 2$	b minus a equals d
$a \times b = e$ $a\ b = e$ $a\ b = e$	$2 \times 4 = 8$	a times b equals e, or a, b equals e
$a\,(a + b) = f$	$2 \times 6 = 12$	a into a plus b equals f
$b \div a = h$ $\frac{b}{a} = h$	$\frac{4}{2} = 2$	b divided by a equals h, or b over a equals h
$a < b$ $b > a$	$2 < 4$ $4 > 2$	a is less than b b is greater than a
$b:a::f:c$ $\frac{b}{a} = \frac{f}{c}$	$\frac{4}{2} = \frac{12}{6}$	b is to a as f is to c b divided by a equals f divided by c b over a equals f over c
$a^2 = b$	$2 \times 2 = 4$	a square equals b
$b^3 = k$	$4 \times 4 \times 4 = 64$	b cube equals k
$\sqrt{b} = a$	$\sqrt{4} = 2$	square root of b equals a
$\sqrt[3]{e} = a$	$\sqrt[3]{8} = 2$	cube root of e equals a

Let us now take the simple case of finding the area of a circle whose diameter we know. Expressed in English the rule is: Multiply the diameter by itself, then multiply the resulting product by 0.7854. The result is the area of the circle. If the diameter is expressed in inches the area will be expressed in square inches. The corresponding engineering

expression is

$$A = 0.7854\,d^2 \qquad (1)$$

where A = the area in square inches,
d = the diameter in inches.

Now, to solve this expression for a particular case, suppose we wish to know the area of a circle nine inches in diameter. We simply substitute for d^2 its numerical value, and perform the indicated operations, thus:

$$A = 0.7854 \times 9 \times 9 = 0.7854 \times 81 = 63.617 \text{ square inches.}$$

Take as another example the formula for the indicated horse power of an engine:

$$\text{HP} = \frac{PLAN}{33{,}000} \qquad (2)$$

where P = the mean effective pressure in pounds per square inch,
L = the length of stroke in feet,
A = the area of the piston in square inches,
N = the number of strokes per minute.

The whole information as to how to determine the indicated horse-power of an engine is given in the above small space, while to write the same in English would require half a column or more of the space at our disposal.

Take the case of an 8 x 10-inch engine running at 100 revolutions per minute under 125 pounds mean effective pressure; here we have:

$P = 125$ pounds,

$$L = \frac{10 \text{ inches}}{12} = 0.833 \text{ feet,}$$

$A = 0.7854 \times 8 \times 8 = 50.26$ square inches,
$N = 100$ rev. per min. $\times\, 2 = 200$.
Then,

$$\text{HP} = \frac{125 \times 0.833 \times 50.26 \times 200}{33{,}000} = 31$$

A reader of engineering literature frequently encounters trigonometrical expressions, and must know how to treat them. They are in reality about the easiest of all engineering terms to deal with, although often proving a stumbling block to beginners because of their peculiar names and unfamiliar looks. They are always used in connection with a given angle, and are called the *trigonometrical functions* of the angle.

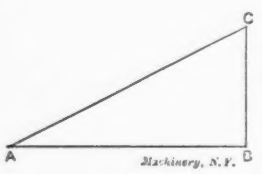


Fig. 1.

They are as follows for angle A in Fig. 1: in the formulas where the sides are given as AC, CB, etc., the length of the side in inches is meant:

TABLE NO. 2.

CB = opposite side, AB = adjacent side, AC = Hypotenuse.

Called.	Written.	Rule.
sine of A	$\sin. A = \frac{CB}{AC}$	opposite side divided by hypotenuse.
cosine of A	$\cos. A = \frac{AB}{AC}$	adjacent side divided by hypotenuse.
tangent of A	$\tan. A = \frac{CB}{AB}$	opposite side divided by adjacent side.
cotangent of A	$\cot. A = \frac{AB}{CB}$	adjacent side divided by opposite side.
secant of A	$\sec. A = \frac{AC}{AB}$	hypotenuse divided by adjacent side.
cosecant of A	$\text{cosec.} A = \frac{AC}{CB}$	hypotenuse divided by opposite side.

Tables will be found in the hand books giving the numerical values for these expressions for all angles. The following is an example of such a table:

Angle.	sine.	cos.	tan.	cot.
10 degrees	0.1736	0.9848	0.1763	5.671
11 degrees	0.1908	0.9816	0.1943	5.144
12 degrees	0.2047	0.9781	0.2125	5.704

There is also a proposition in geometry so commonly used as to demand recognition whenever met. It is called the rule of the squares of a right angle triangle. In Fig. 2 let ABC be a right angle triangle, having angle CBA for the right angle. If we draw a square on each side as shown, the rule says that the area of $ADEC$ is equal to the area of $ABHI$ plus the area of $BCFG$. The rule is usually expressed thus: The square on the hypotenuse is equal to the sum of the squares on the other two sides. Given expression as a formula the rule is:

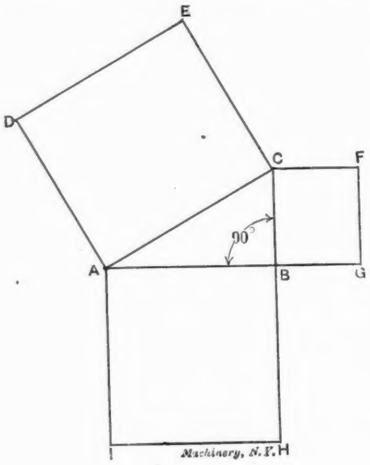


Fig. 2.

$$AC^2 = AB^2 + BC^2 \qquad (3)$$

With the help of this formula and the trigonometrical functions of the angles, we can solve any problems of the right angle triangle, as illustrated in the following example. We have an inclined plane or approach like Fig. 3 to make, and wish to know the angle and the length of the incline, knowing the height BC to be $23\frac{1}{4}$ inches, and the length AB to be 10 feet.

$$AB = 10 \text{ feet} = 120 \text{ inches,}$$

$$tan.\ a = \frac{CB}{AB} = \frac{23.25 \text{ inches}}{120 \text{ inches}} = 0.194$$

Looking in the table of functions of angles we find opposite 0.194 in the column of tangents, that the angle a is 11 degrees. From (3) we have

$$AC^2 = AB^2 + BC^2 = 120^2 + 23.25^2 = 14400 + 540.56 = 14940.56$$

then,

$$AC = \sqrt{14940.56} = 122.3 \text{ inches.}$$

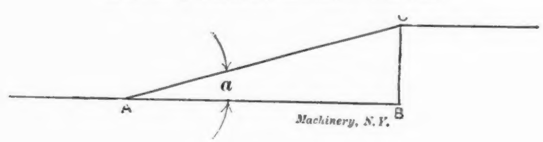


Fig. 3.

Thus we find the angle of inclination is 11 degrees, and the length of the incline is 122.3 inches. A table of square roots will be found in any of the handbooks.

* * *

THE ENGINEERS' DEPARTMENT OF A LARGE HOTEL.

STORRS ELY EMMONS.

In one of the largest hotels in this country, if not in the world, where the number of employees reaches the neighborhood of fifteen hundred, especial attention must be paid to the needs of the mechanics and engineers whose duty it is to keep the house running smoothly in a mechanical sense. For the accomplishment of this end such a hotel must have a thoroughly equipped and systematized engineers' department.

We can readily realize the magnitude of the duties coming under this department when we learn that the hotel in question is provided with eighteen electric and an equal number of hydraulic elevators, an ice plant of far greater capacity than that found in many cities, a large electric plant, pumps for air, water and sewerage, as well as a special set of fire pumps, and a large boiler room equipped with automatic stokers and ash hoists. This department is also responsible for the installation and maintenance of telephones, pneumatic tubes, heating, ventilating, plumbing, clocks, bells and in fact everything of a mechanical nature in the building. This department occupies the entire sub-basement of the hotel and has in its employ over one hundred mechanics, electricians, steamfitters, plumbers, engineers, etc.

The head of such a department must, of necessity, be a man of exceptional executive ability, having a thorough knowledge of all the different branches of engineering. On the pay-roll he is known as chief engineer, but by his associates he is called simply "the chief." Without the liberal

use of systems it would be impossible for him to keep in touch with the men under his charge and with their duties.

In this hotel, which was the first in New York City to organize a mechanical store room, the store room is in no way connected with the steward's but comes under the engineer's department. This department is well stocked with

Date Mar. 14-03 **Order No.** 31426

For Electrician.

Nature Fan. **Location** Room 1672

Remarks Rush.

....

....

Job Completed { Chas. Murdock. 3/14/03 }

Fig. 1. Specimen Order Form.

valves, fittings, plumbers' supplies, waste, engine, dynamo and motor spare parts, electrical supplies and general hardware, in fact, everything that can be called for for rapidly making repairs, also tools and machine parts for the supply of the machine room. The material in the bins of this store room is kept run of by the use of the card system. Each

TIME SHEET.

Date Mar. 14-1903.

Name Chas. Murdock.

Time Number 412

Department Elect.

Job Number.	From	To	Hrs. Min.	Rate.	Total.
31426	8 :05	8 :30	25	.20	.09

Fig. 2. Specimen Time Sheet.

item has a high and low stock number, between which points the amount of material in stock fluctuates. As the stock of any material reaches the low mark, the storekeeper makes out a requisition which, after being O. K.'d by the chief is sent to the steward's department, corresponding to the purchasing department of a manufacturing establishment. This department in turn issues the order on the outside firm from which the material is to be obtained. In the selection of the best places for purchasing materials, etc., the steward's department acts in accordance with the advice and recommendation of the chief. When the material is received, the storekeeper gives the steward's department a receipt for the same, as having been received in good condition, and it is then, after a careful inspection, entered on the cards and placed in the bins.

In the chief's office are kept blue-prints of the entire house showing the exact location of all lights, telephones, fuses, valves, etc. Directly outside of his office door is placed a large blackboard which is divided into sections, each marked with the name of one of the departments under the chief's direction, such as steamfitter, plumber or electrician. Fastened at the top of each of these divisions is a hook such as is used for filing papers. A desk telephone is located at one side of the blackboard and upon this a clerk is in constant attendance. He receives the orders for repair work from different parts of the building and places them on duplicate blank forms, a copy of one of these being shown in Fig. 1. The original of this order is then placed on the hook of the department whose duty it will be to attend to the job. The duplicate of this order is retained on his file for reference. When the job is complete the original order is returned to the department where it supersedes the duplicate on the reference file.

After an order has been placed on one of the hooks it is removed by the first employe of that particular department who is out of work. He marks the location of the job on

REQUISITION.

Date Mar. 14/03 **Job No.** 31426

Time No. 412 **Signature** Chas. Murdock

1	6"	Fan		
10'	#16	Flex. Lamp Cord		
1		Socket Plug		

Fig. 3. Specimen Requisition.

the blackboard, signs his name and the number of the order and goes about the job. In this way it is always possible for the chief to locate his men as well as the work upon which they are engaged.

A daily time card, such as is shown in Fig. 2, is made out by each of the workmen and turned over to the time clerk. This card shows exactly how the man's time has been employed throughout the day. When material is required on a job it is obtained by a requisition on the store room, a copy of one of these being shown in Fig. 3. These requisitions, when filled, are turned over to the cost clerk and, as no material is issued without a requisition, it is a simple matter for the cost clerk to charge the material together with the men's time to the correct piece of work.

In this connection it may be well to state that it is the duty of the cost clerk to distribute the costs of maintaining the engineer's department throughout the departments of the hotel. To accomplish this he makes use of large distribution

DISTRIBUTION.

Department Steward's.

Month March '03

Lights.		Ventilation.		Heating.		Plumbing.		Telephones.	
		31426	12.00						

Fig. 4. Specimen Distribution Sheet.

sheets, the heading of one of which is shown in Fig. 4. These sheets are headed by the name of the department and are divided into double columns, each pair of which is marked with the different appliances which it is the duty of the engineer's department to keep in repair. In one of the divisions

of each column appear the job numbers and in the other the cost of labor and material used upon the respective jobs. A total is taken once a month from these sheets and to this is added a certain percentage of the cost of running engines, dynamos, batteries, etc., which are of general benefit to all departments of the hotel. From these distribution sheets a statement is made out in duplicate, the original going to the auditor's department and the duplicate to all of the departments.

The engineer's department is, of necessity, ready for business every day of the year, and for twenty-four hours of each day, although the night force maintained is as small as possible. While we have not gone into the details of the different methods to any great extent, still the foregoing shows that system in a large hotel as in any other line of business is necessary for running with a minimum loss and producing a maximum output.

* * *

SOME GERMAN OUTSIDE SPRING INDICATORS

DR. ALFRED GRADENWITZ.

The ever increasing pressures used in modern steam engine practice, as well as the successful use of superheated steam, are putting indicators under so severe a strain that any possible errors of design are magnified, and only the best can be used to advantage. Consequently inventors have of late years been examining the indicator critically with a view to a considerable change from the established design, and this point of view has resulted, both in the United States and Germany, in a number of patents for indicators having the spring on the outside of the cylinder.

Indicator investigators concentrated their attention to a great extent upon the piston spring, as the most vital part of the mechanism, and the one most subject to possible error. The sources of possible variation from correct results, in the use of the spring as calibrated, were thought to lie in a too great inertia of the moving parts, piston friction and lateral movement of the spring while being compressed, and changes in the scale of the spring due to its expansion or contraction under varying termperatures.

The "Staus" and "Willner" indicators, recently put on the market in Germany are thought by their inventor to have successfully eliminated these sources of error. The familiar Crosby spring was thought most suitable for these, as in it inertia and momentum are reduced to a low point, the small steel bead by which the spring is attached to the piston being the only mass moved by the wire in addition to its own. Also, the double coiling of this spring effectively counteracts any side thrust. The spring as used on the above-named indicators is seen in Figure 1 to differ from that used on the regular inside spring Crosby indicators only in being inverted, the steel bead being at the top, when the spring is placed on the piston rod instead of at the bottom.

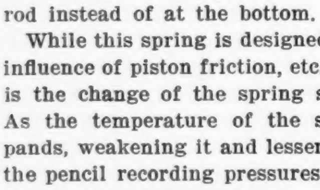


Fig. 1. The Crosby Indicator Spring.

While this spring is designed with a view to diminishing the influence of piston friction, etc., another factor to be considered is the change of the spring scale due to thermic expansion. As the temperature of the spring increases the spring expands, weakening it and lessening its scale; that is, instead of the pencil recording pressures of 40 pounds to the inch of ordinate, for example, the true reading would be perhaps only 39 pounds. This influence cannot be allowed for entirely by calculation, as the temperature of the spring in inside spring indicators is dependent on several variable factors, as the tightness of the piston, the length of time during which the cock is open, etc. Experiments made in Germany are said to have shown that in inside spring indicators departures up to 6 per cent. in the readings have occurred between tests made in the cold or the warm state.

The effect of residual elasticity is much more strongly felt with elevated temperatures than with low. Those who advance the claims of the outside spring indicator believe that by the position of the spring characteristic of that instrument the range of suddenly changing temperature to which the spring is subjected is greatly decreased and with it an important cause of inaccuracy. They also call attention to the rusting effect of the steam or gas to which inside springs are exposed.

The Staus indicator, as shown in Figure 3, is arranged as follows: On the cylinder cover are two steel columns connected above by the spring support and held at the bottom by a nut. The spring is screwed to the support and is acted upon centrally by the contiuation of the hollow piston rod. The ball at the top of the spring has its bearing in a slit at the end of the piston rod, where it is locked by a milled screw. The recording outfit is similar to the one used in the Crosby indicator (ratio 1 to 6), the pencil lever passing around the piston rod symmetrically with a yoke construction so as to prevent any side thrust. The piston with its spring and recording mechanism may be removed as a whole, without being unmounted, upon loosening one nut. The spring may be easily exchanged for another by removing a screw. The height of the atmospheric line on the paper can be varied at will by inserting supporting disks between the head of the spring and its support. According to experiments carried out in Germany, the spring remains cool during the use of the indicator both with saturated and superheated steam.

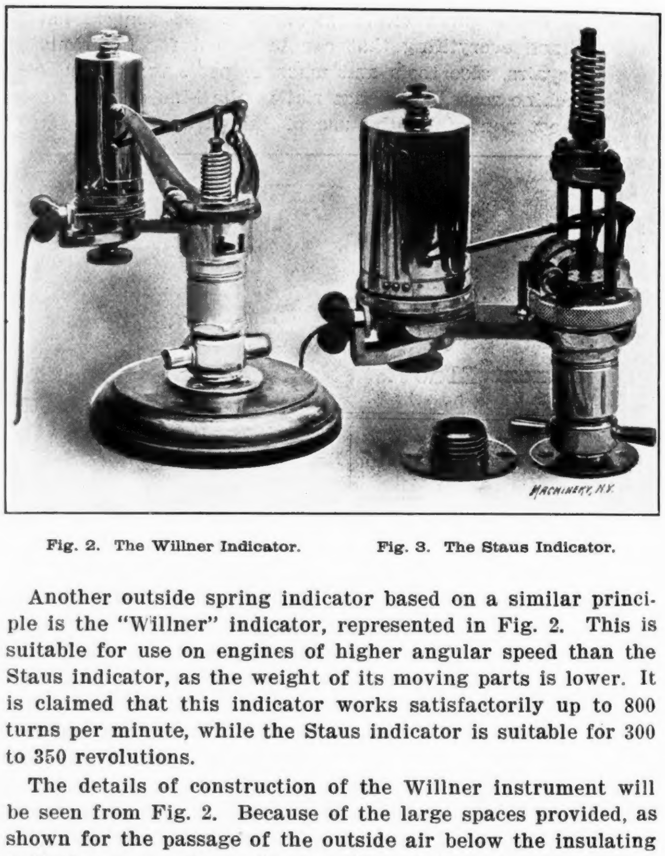


Fig. 2. The Willner Indicator. Fig. 3. The Staus Indicator.

Another outside spring indicator based on a similar principle is the "Willner" indicator, represented in Fig. 2. This is suitable for use on engines of higher angular speed than the Staus indicator, as the weight of its moving parts is lower. It is claimed that this indicator works satisfactorily up to 800 turns per minute, while the Staus indicator is suitable for 300 to 350 revolutions.

The details of construction of the Willner instrument will be seen from Fig. 2. Because of the large spaces provided, as shown for the passage of the outside air below the insulating indicator cover, the spring is said to take a temperature no higher than that of the hand even after prolonged service.

The patented instruments described above are being brought out by their inventor, Mr. H. Maihak, of Hamburg, Germany.

* * *

Pening, riveting, calking, and battering are terms that mean very much the same thing to the lay mind, but there is, of course, a great difference. For example, the difference between pening and battering a piece may be expressed by saying that pening is a systematic striking of the work with the pene of a hammer, so as to cause the metal to flow into a regular shape, and battering may be the same work unskillfully done, the blows being delivered with either part of the hammer, the pene or face, as it may happen, and struck unsystematically over the work as an unskilled workman may direct. Pening may be the same as riveting, as when the end of a pipe is pressed over to fill a groove bored in the face of the flange. Calking and pening may be the same operation, but ordinarily calking requires the use of an intermediate tool between the hammer and the work.

HARD LINES FOR DRAFTSMEN.

H. E. WOOD.

Don't make uneven lines.
Don't fail to take notes.
Don't make pale blueprints.
Don't upset your ink bottle.
Don't roll a drawing too tight.
Don't fail to trim each tracing.
Don't use a ruling pen that cuts.
Don't go to sleep over your work.
Don't worry; it injures your work.
Don't fold tracings; it injures them.
Don't make a dirty looking drawing.
Don't make figures out of proportion.
Don't make a mistake adding fractions.
Don't work on wrong side of the paper.
Don't fail to pick up "dropped tacks."
Don't try to work with a blunt pencil.
Don't put the wrong number on a drawing.
Don't give two drawings the same number.
Don't take orders over your chief's head.
Don't be reluctant about making sketches.
Don't put your pens away with ink in them.
Don't waste material; it is very expensive.
Don't take orders from anyone but the chief.
Don't leave tacks, point upward on a board.
Don't leave out important lines on a drawing.
Don't keep your tracing cloth in a damp place.
Don't give out a drawing until it is complete.
Don't use instruments with lost motion in them.
Don't be backward about taking responsibility.
Don't make center lines heavier than main lines.
Don't make unnecessary tack holes in a drawing.
Don't draw a line unless you know what it is for.
Don't be "earless"; it may land you in the street.
Don't pull out thumbtacks with your finger nails.
Don't hold your instruments in an awkward position.
Don't crowd the different views; it shows bad taste.
Don't cover a drawing with unnecessary measurements.
Don't get angry if you are asked to change a drawing.
Don't be thoughtless, or you won't be very successful.
Don't punch your scale full of holes with the dividers.
Don't fail to give every detail necessary consideration.
Don't let tracing cloth get wet, for then it is spoiled.
Don't keep a lot of unnecessary drawings laying around.
Don't forget that it is bad practice to do too much erasing.
Don't lose sight of the fact that sections are very important.

Don't put measurements on a drawing in an improper manner.

Don't try to do the chief's work, until you can master your own.

Don't leave your mistakes for others to find; find them yourself.

Don't ask unnecessary questions of the chief; reason them out yourself.

Don't fail to consider the convenience of the machinist whenever possible.

Don't make the lines too faint on a tracing, for they make poor blueprints.

Don't make a drawing unless you can see in your mind just what the piece looks like.

Don't put a measurement on a drawing that won't make up to correspond with the total.

Don't tear up sketches as soon as the drawing is made; they serve as proofs, sometimes.

Don't forget to use round-point pencils for figures and letters, and flat points for lines.

Don't forget that the patternmaker should be considered sometimes, for he has to work to your drawings.

Don't put figures on a drawing unless you know what the consequences will be when some one else works to them.

Don't forget that the machinist will always try to blame the draftsman for his own mistakes if he possibly can do so.

Don't fail to be prepared with reference data for proportions, etc., such as are found in MACHINERY data sheets.

Don't forget that a mechanical draftsman should also be a machinist, and have a first-class idea of foundry, pattern, and blacksmith shop work.

Don't forget that the point of a draftsman's pencil is the turning and starting point of many long roads; so be careful how you handle it.

Don't fail, when dimensioning a drawing to go into the machine shop, to give all measurements as far as possible, from some main or general surface, as it is easier for the machinist to work from.

* * *

A SET OF ACCURATE TEST AND INSPECTION GAGES FOR SMALL DUPLICATE WORK.

JOSEPH V. WOODWORTH.

In Figs. 3, 4, 5, and 6, is illustrated a set of four tests and inspection gages of interesting design, possessing adaptable features which may be used to advantage in similar gages for verifying measurements of small, accurately machined steel parts. All four of the gages are graduated to indicate variations down to .0005 inch.

The work for which the inspection tools were utilized consisted of two parts of tool steel, which were machined to

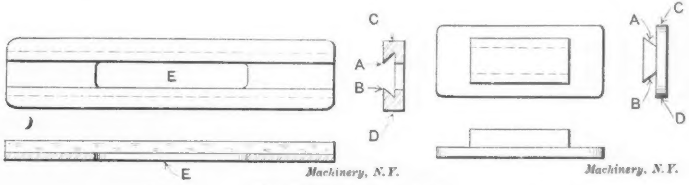


Fig. 1. The Pieces to be Gaged. Fig. 2.

extremely accurate dimensions, and were required to fit or assemble together perfectly. These two parts are illustrated in Figs. 1 and 2. The part shown in Fig. 1, as may be seen, has a dovetailed channel milled in one side for its entire length, and has the web punched out to form a hole at *E*. This part, after all milling operations had been concluded, was hardened and tempered; the temper being drawn down sufficiently low to allow shaving the dovetailed surfaces to finish sizes with "Novo" steel cutting tools, after which the sides and edges of the part were ground to limit sizes with a cup shaped emery wheel in a specially constructed grinder. It was after this operation, that the two gages illustrated in Figs. 3 and 4, were used to determine the degree of interchangeability attained in the parts through the various concluding mechanical operations.

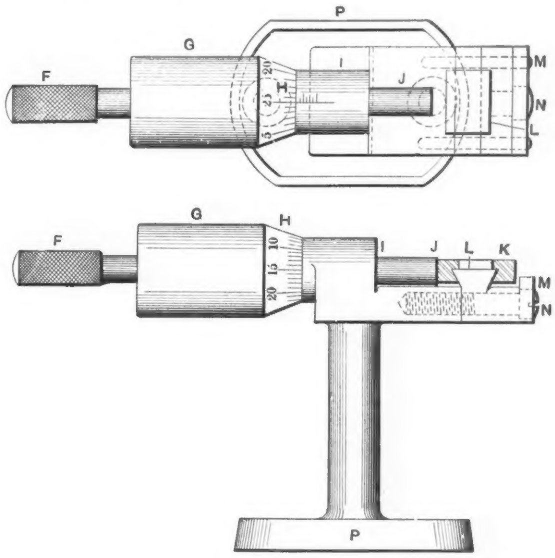


Fig. 3. Micrometer Gage for Thickness of Side of Fig. 1.

The micrometer gage shown in Fig. 3, was used to determine the width of the part from the dovetail to the edges, as is indicated in the lower view, in which is shown a cross-section of the work in position on the dovetail locating piece *L*. After gaging one side, in the manner outlined, the other side is inspected by simply reversing the work on the dovetail locator. The drawing of the gage is so clear and self-explanatory that a detailed description of construction will be un-

necessary. *F, G, H, I* and *J* comprise the micrometer portions, *L* is the locator, *M* is the block upon which it is formed, *N* the fastening screw and dowels which fasten and locate it in position on the body of the gage, *P* is the base and *O* the stem of the gage. Two flathead screws were used to fasten the gage to the work bench. All locating and test parts of the gage were hardened and carefully ground and lapped.

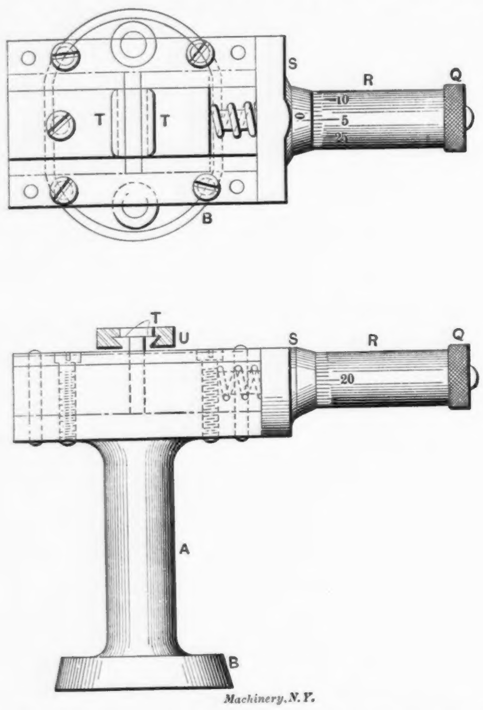


Fig. 4. Micrometer Gage for Width of Channel of Fig. 1.

The micrometer gage illustrated in Fig. 4, was used for measuring the width of the dovetail channel in Fig 1; that is, the distance between points *A* and *B*. This gage consists of base *B*, which is fastened to the work bench when in use, stem *A*, two slides *T T*, which have raised angular faced projections which conform to the angle of the dovetail in the work, and the micrometer portions *S*, *R* and *Q*. The manner in which the gage is used to determine the width of the dovetail is clearly indicated in the lower view of Fig. 4, a cross section of the work being shown in position. By revolving the barrel, by means of the knurled end *Q*, the slide *T*, at the right, moves back, to allow removing or locating the work; while the revolving of the barrel in the opposite direction causes the slide *T* to move out and expand to the full width of the dovetail channel in the work. The other slide, *T*, is fixed. A light, spiral spring, which is shown in the plan of Fig. 4, assists the slide to move back readily when the barrel is revolved and the work is to be removed or located.

It will be noticed that both gages described in the foregoing, are simple in design and of comparatively inexpensive construction, when the rapidity with which the work may be handled and inspected in them, and the required accuracy of the tests, are considered. In case of wear, in any of the precision parts of either gage, the error may be rectified by simply making suitable adjustments, provision for which may be seen in the drawings; thus no difficulty is experienced in maintaining the accuracy of the tools.

In Fig. 2 is shown the dovetail slide which is required to assemble on the part shown in Fig. 1. This piece is also made of tool steel, is accurately machined all over, shaved on the dovetail surfaces, *A* and *B*, and then ground on the back. It was for verifying the interchangeability of this part that the decimally graduated inspection gages shown in Figs. 5 and 6, were used for determining the exact amount of variation in the parts from the inner edge of the dovetail, *A* and *B*, to the ground edges of the body portion at *C* and *D*.

The gage consists of a flat cast iron base, *F* cored at *G* and equipped with four steel legs, *K K K K*. *W W* is the test and gage portion proper, and consists of a dovetail piece of tool steel which is fitted and gibbed into a channel in *F*, as shown, by gib *P* and screws, *O O*. This piece is milled away in the center so as to provide a clearance way for the pointer *J*, which is pivoted on a small pin at *N*. The projecting end *Q* of *W* is hardened, ground, and lapped to conform to the angle of the shaved dovetail surfaces of piece Fig. 4; it is also drilled longitudinally in the center of *Q*, to accomodate the plunger, *R*, which is fitted to "float" in the reamed hole. The rounded end of *R* at *S* rests against the edge of the pointer. Pin *T* in the end of the plunger prevents it from getting away, and the light spring *M*, serves to keep the plunger out when the work is not against it. Knurled head screw *U*, spanned by yoke *V*, and screwed into *F*, is utilized to correct any inaccuracies in the precision parts which occur through use and wear. The graduations for reading the tests are at *I*, at the extreme end of *F*, with the end *J*, of the pointer matching them. When the pointer registers at *O*, the work is up to the requirements; if it points to 5 at the left the work is considerably too large; if it points to 5 on the right the work is too small.

In using the gage it is placed on the bench before the inspector and the dovetail slide Fig. 2 is located on it with one dovetail edge resting against the angular face of *Q*, and the edge *D* resting against plunger *R*; thus, upon the inspector pressing down and in upon the work the edge forces the plunger *R* inward, and therefore causes the pointer to register the reading at *I*. A spanner of small diameter drill rod at *H H*, acts as a guard for the pointer.

In Fig. 7 we have another decimally graduated test gage, used in the inspection of part Fig. 2. It is used to determine the exact width of the dovetail portion, from *A* to *B*. A base of cast iron is machined to accomodate the hardened and ground angular faced pieces *I* and *F*. Piece *I* is fastened and

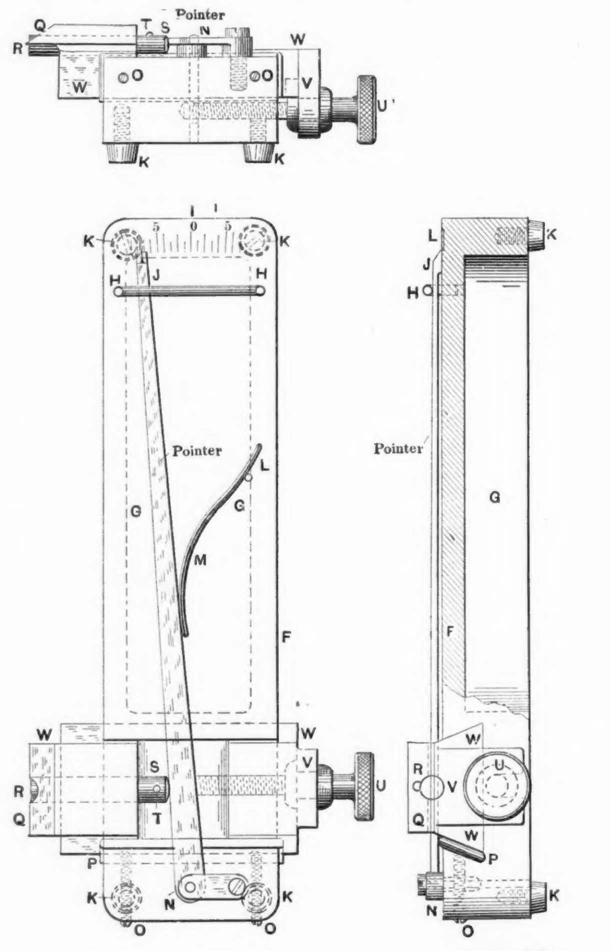


Fig. 5. Micrometer Indicator for Location of Dovetail of Fig. 2.

located by screws *KK* and *JJ*, while piece F is left free to slide in the holder. It will be noticed in the plan view of this gage that the dovetail edges *G* and *H* of *I* and *E* respectively are at an angle of a few degrees with the dovetailed channel in the base; thus as the slide *F* is pushed forward the space between *G* and *H* decreases; and as it is pulled back in the opposite direction the space increases; therefore by simply en-

tering the work so that the dovetail part rests between *G* and *H* and then pushing the slide forward until the work is clamped or held tightly, a reading may be had instantly by noting the relation of the graduated line with zero point *O* at *P*.

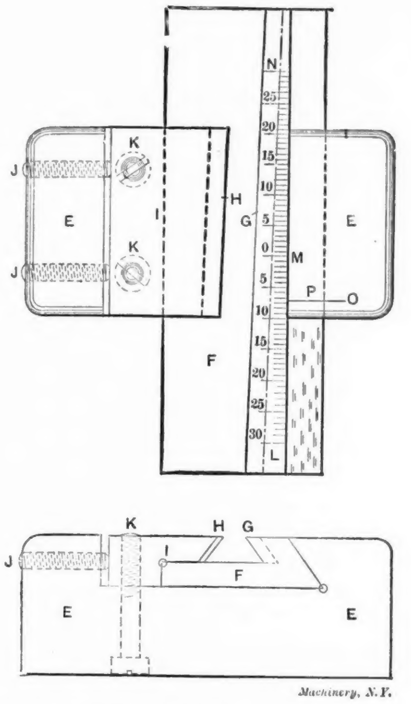


Fig. 6. Micrometer Gage for Dovetail of Fig. 2.

Although gages of the types described in the foregoing may appear quite simple and easy to construct, considerable skill and care are necessary in the grinding, lapping and graduating, in order to produce reliable precision instruments for the inspection of accurate work.

* * *

RELIEF OF TAPS.

H. D.

In the manufacture of taps one of the most particular and serious questions arising is how to give a proper relief to different kinds of taps. Any one familiar with the making of taps knows, of course, that a hand tap should be given a different relief than, for instance, a pipe tap. For this reason the different kinds mostly used have been treated separately, each kind by itself.

Hand Taps.

The old, and up to some time ago universally used way, was to give all the teeth a relief in the angle of the thread, *i. e.*, the heels of the teeth were made of smaller diameter than the diameter measured over the cutting edges (as shown in the end view of Fig. 2). However, this has been found to be wholly unnecessary, and taps of this kind are now made without any relief whatsoever in the angle of the thread; but the top of the thread of the *chamfered part only* is slightly relieved. To further improve upon the cutting qualities of the tap, it should be made smaller in diameter toward the shank than at the point. This difference in diameter should, of course, vary for different diameters, and the limits in variation of size permitted must, of course, also be taken into consideration. It may be said that in general practice it is answering the purpose if the tap is about 0.0015 inch smaller at the shank end of the thread for taps up to ½ inch diameter, and from 0.002 to 0.003 inch smaller at this end than at the point for taps from ½ up to 2 inches diameter. It may be added that although this is an essentially good point in tap making, most manufacturers do not make their taps that way, probably because it would increase the expense in the manufacture, and require greater care in making.

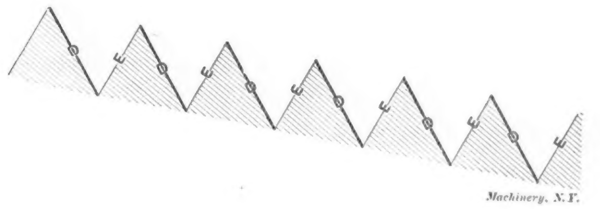


Fig. 3. Section of Cutting Edge of Taper Tap.

Another improvement upon a hand tap, seldom seen in taps manufactured for the market, is to give to the angle of the thread a relief in the center of the land, as is shown in Fig. 1. The reason for so doing is obvious. The tap gets the same support along its periphery as if not relieved in the angle of the thread, because it retains its bearing at the heel of the thread, but as can be clearly seen a good portion of the friction is eliminated.

Taper Taps.

In order to fully explain what is referred to as a taper tap in the following, it may be well to say that such a tap has the part of the thread nearest the shank larger in diameter than at the point of the tap, as, for instance, a pipe tap. This is mentioned on account of the fact that the first tap in a set of hand taps is commonly, but not properly, referred to as a "taper tap," and it would perhaps cause confusion if special attention was not called to the correct meaning of the term "taper tap."

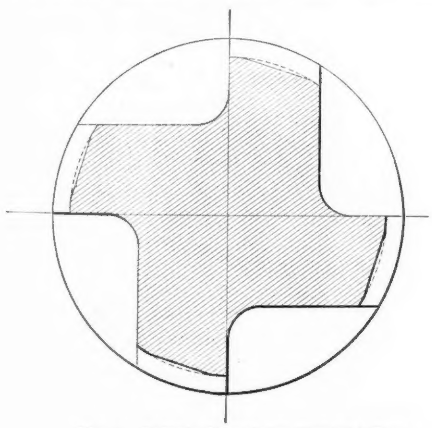


Fig. 1. Relief of Land for Straight Tap.

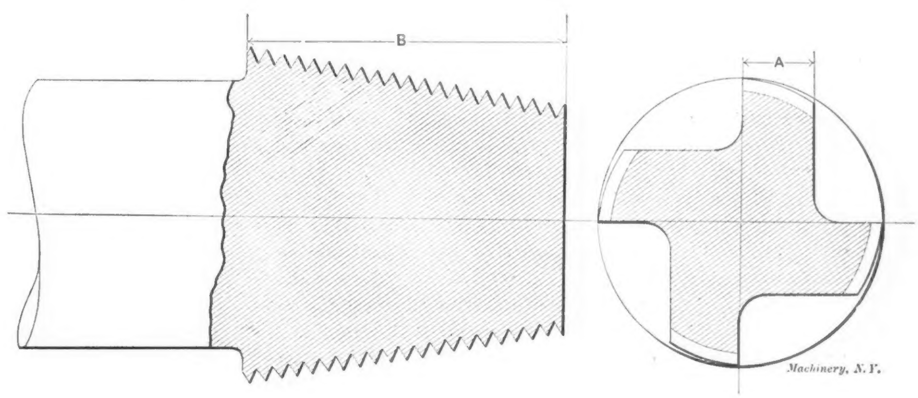


Fig. 2. Relief for Taper Tap.

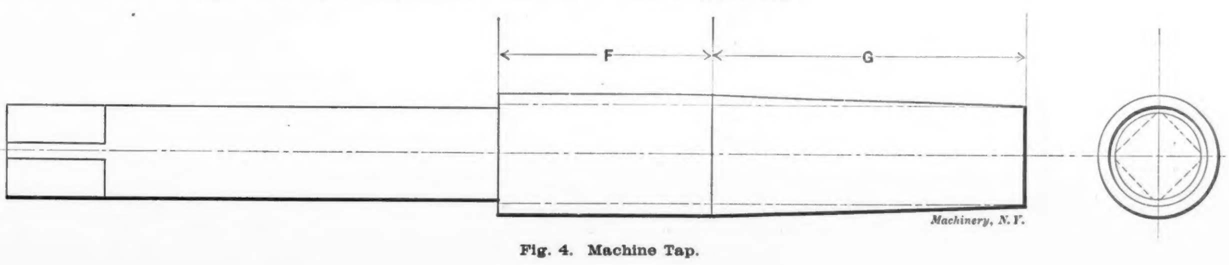


Fig. 4. Machine Tap.

It is obvious to any one considering the matter that a taper tap (Fig. 2), relieved in the same manner as a hand tap described, would refuse to cut altogether. A tap of this kind should invariably be given a relief the entire width of the tooth *A*, and the full length of the thread *B*. It is also to be noted that the greater part of the relief should be given on the side *D* of the thread, to lessen the friction and the resistance while cutting (see Fig. 3). That such a result will be obtained will be perceived after a careful consideration, because the pressure on the thread of the tap that is created by the cutting process will all come on this side of the thread, and if then relieved properly so as to only permit the *cutting edge* to come in contact with the material to be cut, it is obvious that the friction is reduced to the smallest possible amount, at the same time as the keenness of the cutting edge is increased.

Machine Taps.

Under this heading are treated taps used for threading nuts in special nut-tapping machines. Such taps are subjected to very hard usage, and must therefore be made in a special way, and with special care.

A tap of this kind has a part, *G*, tapered on the outside of the thread, and a straight part, *F*, as will be seen from the cut. The length of the tapered part should depend upon the material the tap is to be used upon, and also upon the length of the nut to be tapped (the longer the nut, the longer the part of the tap tapered on the outside should be). This part, *G*, ought to be relieved both at the top and in the angle of the thread, *i.e.*, the diameter measured over the heel should be smaller than the diameter measured over the cutting edge.

The straight part, being nothing but the sizing part of the tap, should be left without relief, or if any should be given, it ought to be very slight, so as to permit the tap to retain its size all the longer.

Ordinary die taps can also be made in the same way as a machine tap.

Hob Taps.

As these are generally used for "burring" and sizing, and have little or no cutting to do, they are mostly made with no relief at all. An exception from this might be made in making a taper hob tap which might be slightly relieved on the same side of the angle of the thread as is shown at *D* in Fig. 3.

* * *

VISITING SHOPS.

ENTROPY.

What a variety of people wander into a shop in the course of time. We get all kinds from the proprietor of the big shop over in town down to the pretty young misses who come in to see the wheels go round, and who go away wondering why they cover up all the new castings with dirt, and why men who just stand and watch a machine go don't dress like bank clerks. Once in a while a few of the "Tech" boys come in who watch us cutting a pinion for a while and then call to another fellow to come over and see us fluting a reamer. Their professors also come once in a while, and they are more fun than the boys in exactly the proportion that they try to be more serious. It often occurs to me that it is a really wonderful thing that a college or technical man ever succeeds in bucking the world at all after being mewed up for years with instructors whose ideas of what is going on in the world are as vague as those that we see here. I suppose, though, that if a boy has brain and vitality enough to go through a technical course he is capable of almost anything when he gets out and has a chance.

Then, there is the new agent on the road, selling paint or oil or some other thing distantly related to the trade. He adds to our fund of humor by selling us white asphalt, four-sided three-square files and other novelties passed down from Noah, but leaves us without adding to our store of knowledge.

Next is our friend who advises us what machine tools to buy and clings to us till he gets his commission, unless in the meantime he gets to criticising our pet equipment; then we turn him down like a lump of lead. Why, I have known of a large and prominent firm turning down an equally important make of turret lathe simply because the builders of the latter advised making a certain simple piece in a reverse order from what their superintendent had been in the habit of doing. But isn't it curious what instincts different occupations develop? Did you ever see a blacksmith that didn't scowl? He gets it looking at the fire and carries it home and passes it along. Did you ever see a lawyer or a teacher that wouldn't criticise anything and everything just for the sake of hearing himself knocking? It's part of their business. If they for a moment should forget themselves and pat some one on the back and call him a good fellow, where would their jobs be? Did you ever see an old salesman who was narrow or bigoted? They couldn't stay on the road if they were; they either have to reform or drop it. Daily and hourly intercourse with men who have dug out fortunes from the cold side of the world puts a man where he can appreciate the good that men do and where he can see something of the relative value of the ten commandments, and while the traveling man may have his own favorite vices, he will average more real men to the thousand than most of those who run him down.

Most shop owners, or the men who run shops, are like a lecturer, or a minister, or an actor—they like to see people who are interested in what they are doing and who are willing to make their interest manifest. Every one of them either has ideas of his own or else is getting out of business as fast as he can, so you don't want to see his shop. These shop men are all looking out for ideas, but an idea is like a trout; you want to fight for it and catch it yourself. So don't go visiting shops and handing out your ideas of how they should be run the same day. Remember that advice is usually worth what it costs, and if you have any to spare store it for a rising market. And remember, too, that comparisons are odious, and don't tell him how some one else does. He may be too polite to tell you that the other fellow is a back number, or he may get on his ear, as I did once. One of our neighbors came in and looked at some lathe aprons we were building. He remarked in a disdainful tone that he didn't build them that way. To which I replied that I knew he didn't, because he built his exactly like R———'s, and I had always wondered which one stole it!

If you want to get in your man's best graces, look around and comment favorably on something that he has not pointed out; then you are sure to see everything that he can show you, and he will send you away with a "pleased-to-have-met-you" smile that will last clear home. Visiting shops is an art in itself, and the chief art of it is to get in the good graces of the man who is showing you around. You will inevitably get some wrong impression, so don't tell anything unfavorable of a shop to any one before making at least a second visit. A frank avowal of your business and of what you wish to see is a better introduction than all the letters you can carry, for the vast majority of shop men are used to judging others, and they like the tub that stands on its own bottom.

* * *

A description of an ice-making plant which recently appeared in a daily paper, contains the following statement: "Ice made in this plant has good lasting and refrigerating powers." At first thought this remark looks somewhat peculiar, since we are naturally inclined to believe that ice is ice, whether it be chopped from an iceberg or made in a refrigerating plant. There is some basis, however, for making a difference between the lasting qualities of artificial and natural ice, although the newspaper reporter may not have understood just what that basis is. When ice is made under the can system, the freezing of the water leaves a spongy core in the center of the cake. This means that, as compared with a block of solid ice of the same dimensions, the superficial area will be the same, while the weight of the artificial block will be less, so that for a given weight the latter form of ice will melt more rapidly. As to its "refrigerating qualities," it will probably take the same number of heat units to melt a pound of ice under the same conditions, no matter how it was frozen. The ice plant referred to is one erected recently at Niagara Falls, in which electric power is used exclusively. The water is frozen in large tanks, thus avoiding the spongy core referred to above.

CONDENSING PLANTS FOR HIGH VACUUMS.—1.

The main hindrance to the realization of the many claims for the steam turbine appears to be the condensing system with which it must be equipped to produce the best results that the turbine is capable of. The turbine by itself is simple, apparently is durable, economical of steam, and requires but little attendance. Add to the turbine, however, a condensing plant consisting of a surface condenser large enough to allow a vacuum of 28 or 29 inches, an air cooler attached to the condenser, a circulating pump large enough to handle twice or more the usual quantity of cooling water, wet and dry vacuum pumps, the latter being a compound or two-stage pump in some cases, and the engines or motors to drive this apparatus, and one has a really formidable steam plant. These various auxiliaries must be maintained at the highest state of efficiency and all joints kept tight to avoid leaks. The condensing apparatus in turbine plants has generally given more trouble, required more attention, and has been a greater source of expense than the turbine itself, and in spite of the fact that the latter is a new type of apparatus which cannot be considered to have been perfected. These difficulties, however, have often been due to lack of careful lay out of plant and will grow less with the more general substitution of motors for reciprocating engines and of rotary for reciprocating pumps for the auxiliaries.

The reason why it is so important to have an elaborate condensing plant is that the turbine is capable of taking full advantage of high vacuum, by expanding the steam down to the lowest pressure that can be attained in a condenser, thus making use of the heat energy represented by this difference of pressure. In a turbine there is no restriction to the number of times that steam may be expanded, except that the passages must be large enough to accommodate the increased volume of the steam at the low pressures.

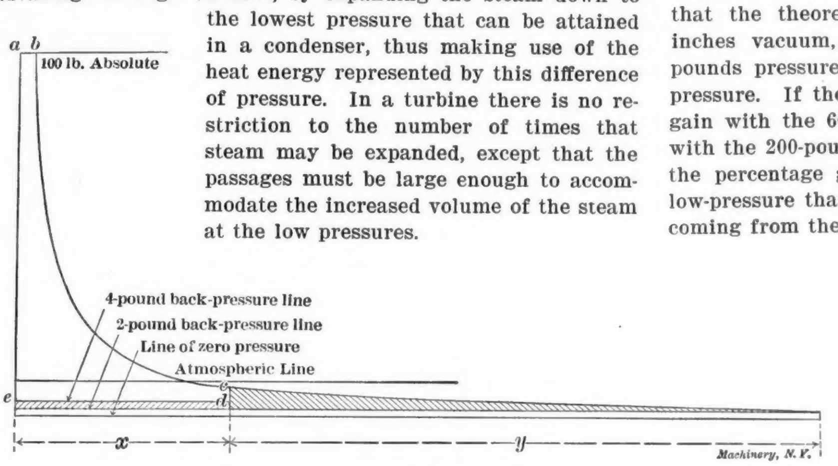


Fig. 1.

In a steam engine the gain from increasing the vacuum above 26 inches is so slight as not to warrant the extra expense of the large condensing apparatus required. A compound engine with the usual cylinder ratio of 4 to 1 will expand the steam from ten to fifteen times. The volume of one pound of steam at 150 pounds pressure, absolute, is 3 cubic feet. In expanding 15 times, or to a volume of 45 cubic feet, the terminal pressure in the low-pressure cylinder would be between 8 and 9 pounds, assuming cylinder condensation to be balanced by re-evaporation. When the exhaust valves open, therefore, the pressure would drop to that of the condenser, and the only effect of a vacuum higher than that represented by the eight or nine pounds pressure in the cylinder is to reduce the back pressure against the piston during the return stroke.

In high-ratio compound engines, having a cylinder ratio of about 7 to 1, steam is expanded 20 or 25 times and more heat energy is theoretically converted into mechanical work than in the previous case, which would make the engine more efficient were it not that the losses from condensation and friction are enough greater to nearly or quite balance whatever gain there may be. If the attempt is made to carry expansion too far in a steam engine, the low-pressure cylinder, valves and passages must be abnormally large and will offer a great deal of frictional resistance. It will be evident that under such conditions a point will be reached in the low-pressure cylinder where the pressure of the steam will not be sufficient to overcome the frictional resistances, to say nothing of doing useful work, and the expansion of the steam beyond this point will therefore be a dead loss, and the engine, instead of giving up power, will require power to drive it.

Table I. gives the volume of one pound of steam corresponding to different "vacuum" pressures and indicates how impossible it is to utilize these low pressures in the steam engine.

TABLE I.

Absolute Pressure.	Vacuum, Inches.	Specific Volume.
½	29	636
1	28	335
2	26	174
3	24	118
4	22	90

To expand steam from 150 pounds to 1 pound absolute, or to 28 inches vacuum, would mean that the volume must increase 111 times. To carry the expansion to this point in a compound engine, the ratio of the cylinders would have to be about 33 to 1; that is, the diameter of the low-pressure cylinder would be 10½ times that of the high-pressure cylinder—quite an impracticable figure.

The steam turbine, on the other hand, needs all the expansion that can be given the steam, and is able to derive full benefit from it. An idea of what the gain is from the use of a high vacuum can be obtained by referring to a few calculations. Konrad Anderson, in a paper upon steam turbines in the Transactions of the Institute of Engineers and Shipbuilders of Scotland, 1902, compares power values for steam expanding from 60 and 200 pounds, respectively, and finds that the theoretical gain in running condensing, with 25 inches vacuum, is nearly 100 per cent with steam at 60 pounds pressure, and 50 per cent with steam at 200 pounds pressure. If the vacuum be then increased to 28 inches, the gain with the 60-pound steam will be about 22 per cent and with the 200-pound steam about 18 per cent. This shows that the percentage gain from running condensing is more with low-pressure than with high-pressure steam and that the gain coming from the last few inches of vacuum is relatively much more than from the first few inches.

This latter fact has been brought out in a striking manner by Ernest N. Janson in an article upon steam turbines in the *Journal of the American Society of Naval Engineers*, 1903. He shows that with the initial and terminal pressures in the same ratio, the kinetic energy of the steam in flowing from a higher to a lower pressure is nearly the same, without regard to what the initial pressure is. For example, supposing the initial pressure to be 105 pounds and steam to expand to one-third this pressure, or to 35 pounds, he finds the kinetic energy developed by the steam to be only 10 per cent more than when expanding from 3 pounds to 1 pound. The figures are as follows:

$\frac{p_1}{p_2} = \frac{105}{35} = 3$; velocity $= 2{,}050$ ft. per sec.; H. P. per lb. of steam per hour $= 0.033$.

$\frac{p_1}{p_2} = \frac{15}{5} = 3$; velocity $= 1{,}933$ ft. per sec.; H. P. per lb. of steam per hour $= 0.029$.

$\frac{p_1}{p_2} = \frac{3}{1} = 3$; velocity $= 1{,}790$ ft. per sec.; H. P. per lb. of steam per hour $= 0.025$.

Fig. 1 illustrates the expansion of one pound of steam from an initial pressure of 100 pounds to the pressures indicated, and illustrates the difference between the way in which an engine and a turbine are able to benefit from a high vacuum. It shows the work done both before and during expansion, as in an indicator diagram. The section of the diagram marked *a-b-c-d-e* represents that part of the energy of the steam that might be converted into work by a condensing engine operating against a back pressure of four pounds, or a vacuum of about 22 inches. When the exhaust valve opens, the pressure drops from point *c* to point *d*. If the vacuum were increased to 26 inches, the gain in power would be due simply to the reduction in back pressure represented by the shaded portion

having the length x on the diagram. This, it will be noticed, is but a small percentage of the total area of the diagram. In the turbine, however, it is different, since expansion can be carried to the 26-inch vacuum pressure within the turbine itself. The turbine is able to utilize the toe of the diagram, indicated by the shaded portion y, in addition to shaded portion x, while the engine is unable to turn to any account the energy represented by the toe of the diagram.

In Fig. 2 are shown the essential features of a turbine condensing apparatus as built by The Henry R. Worthington Co., of the International Steam Pump Co. Steam enters at the top of the condenser, and is distributed over the tube surface by baffle plates, while the condensed steam drops down into the hot well at the bottom, where it is discharged by a rotary pump. This pump requires neither valves nor floats, and is not subject to vapor binding, as are reciprocating pumps. The capacity of the pump is such that it runs ahead of the supply, so that the suction pipe is never full; but the discharge pipe is always full, and the water presses back against the pump; but as long as the latter is in motion it cannot pass back into the condenser.

The air is removed by a rotative vacuum pump which connects with an air cooler or auxiliary condenser. The vapor and air from the condenser passes through this air cooler, through the tubes of which cooling water circulates, and the temperature and specific volume of the air and vapor are

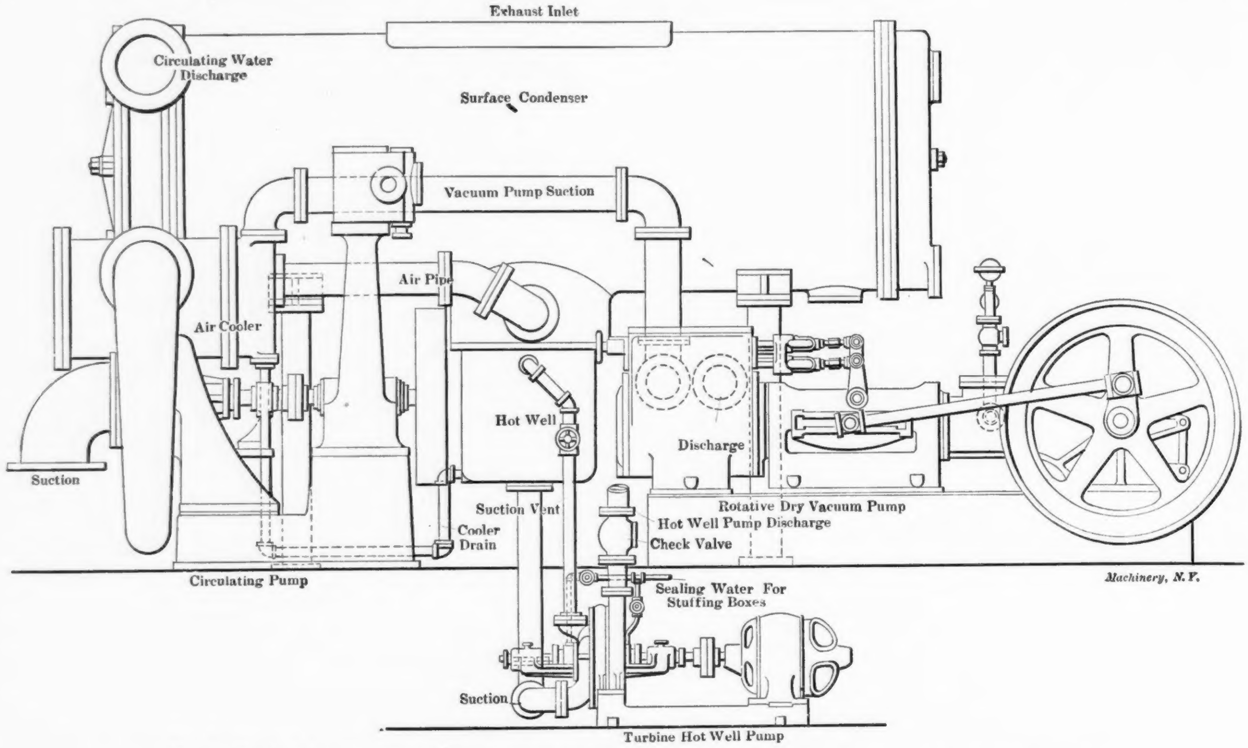
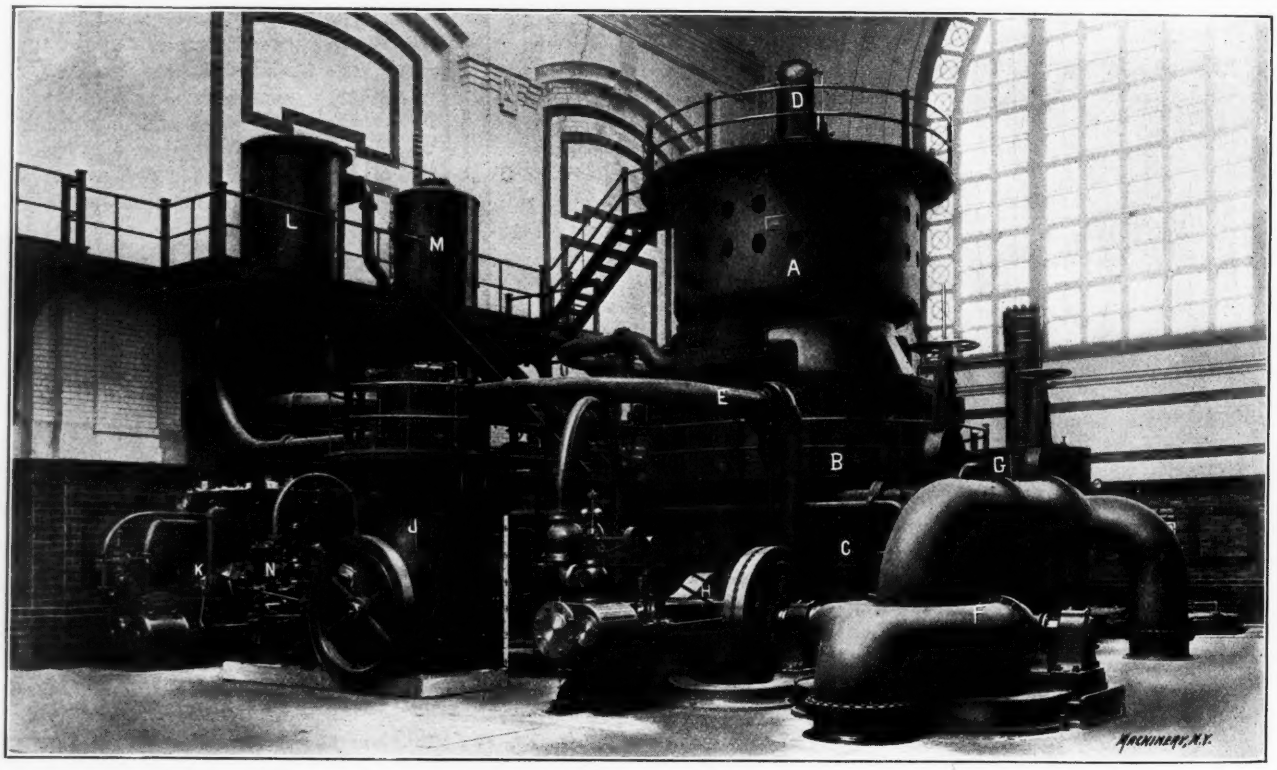


Fig. 2. Typical Arrangement of Worthington Apparatus, with Air Cooler and Wet and Dry Vacuum Pumps.



A.—Generator.
B.—Turbine.
C.—Condenser.
D.—Governor.
E.—Steam Nozzles.
F.—Circulating Pump.
G.—Accumulator.
H.—Engine driving Circulating Pump.
J.—Vacum Pump.
K.—Boiler Feed Pump.
L.—Oil Tank.
M.—Feed Water Heater.

FIG. 3. 500 K. W. CURTIS TURBINE AND AUXILIARIES.

thereby reduced, which enables the vacuum pump to maintain a better vacuum. A rotary circulating pump driven by motor or engine is used for the cooling water.

Fig. 3 is from a photograph of a 5,000-kilowatt Curtis turbine and condensing apparatus installed at the new station of the Edison Electric Illuminating Co., Boston, Mass. In this case, the condenser is built into the base of the turbine, and forms a part of the unit, while the auxiliaries are placed on the same floor level as the turbine itself, where they are more accessible. This illustration gives an excellent idea of the quantity of apparatus required to keep the plant in operation, since the feed pumps, heater, oil tank, and accumulator for supplying the hydraulic pressure that must be maintained under the step bearing of the turbine, etc., are all grouped about the turbine, in addition to the condenser auxiliaries.

In Fig. 4 is a plan view of an Alberger surface condenser and apparatus designed to be applied to a 5,000-kilowatt Parsons turbine. The Alberger condenser is a counter-current condenser, and does not require the use of a separate air cooler. The exhaust enters at the bottom and passes upward over the tubes. The cooling water enters at the top and passes downward, back and forth through the tubes. The air and vapors rising to the top of the condenser are therefore cooled by the incoming cold water and the condensed steam which trickles down into the hot well comes in contact with the surfaces cooled by the warm water as it leaves the condenser, and it is thus possible to maintain a high hot-well temperature without difficulty. A two-stage vacuum pump is employed, one cylinder of which draws the air or vapor from the condenser and delivers it to the other cylinder, which in turn forces it out against the pressure of the atmosphere. Space is saved and the apparatus simplified in this installation by using the same engine to drive both the circulating pump and the vacuum pump, and it will be noted that the condenser and its pumps require just about the same area as the turbine itself, while, by setting the condensing apparatus in a pit, it rises only to the top of the turbine.

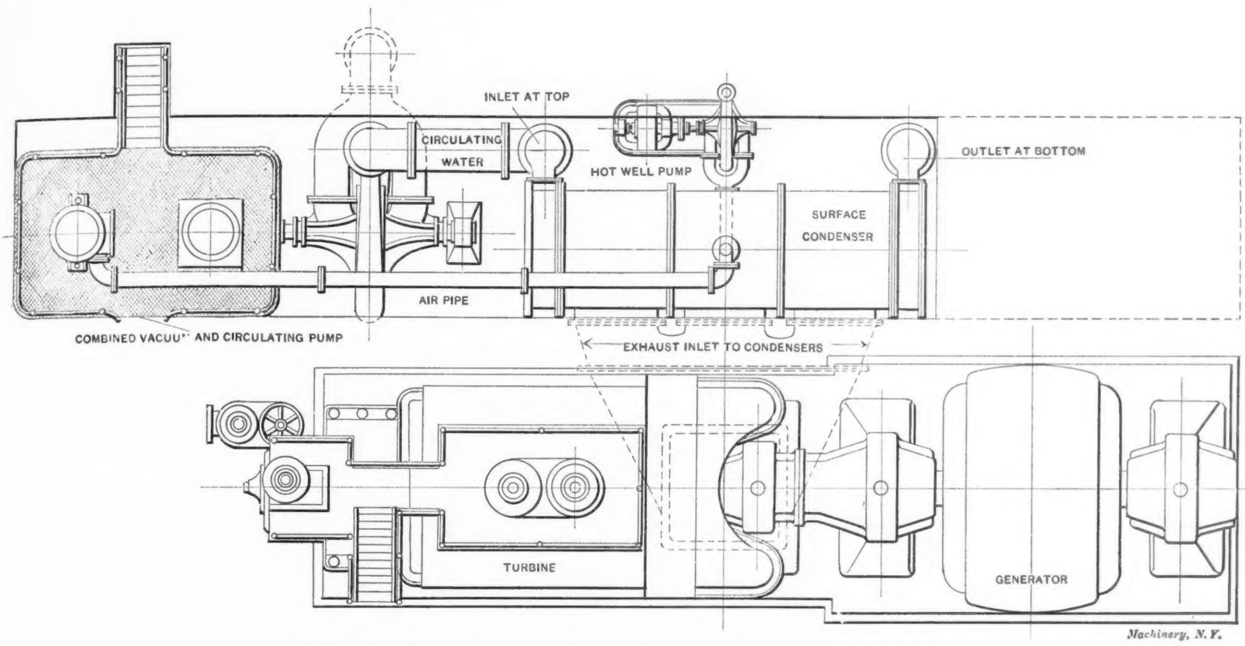


Fig. 4. Plan View of Parsons Turbine and Alberger Condenser and Auxiliaries.

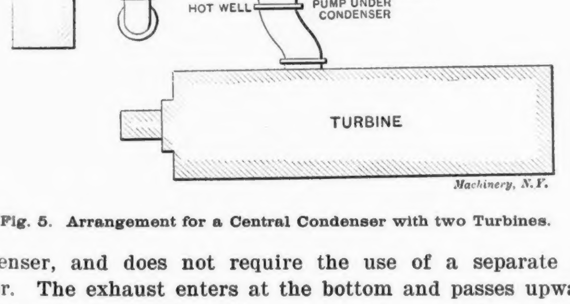


Fig. 5. Arrangement for a Central Condenser with two Turbines.

Where more than one turbine is installed in a station, space may be saved by using one large condenser for two or more turbines. Fig. 5 is a plan in outline indicating the relative area occupied by two 400-kilowatt Parsons turbines, with one condenser placed between them. This plan can even be extended further and a central condensing system employed for several turbines, similar to the condensing plants now used in many of the large power stations; but it has the objection of possible leaks in the long pipe connections required.

* * *

A PLEA FOR BETTER SURROUNDINGS IN THE FOUNDRY.

One great drawback to success in obtaining young men of high standard for our foundries is the environment of the foundry itself, known as it is to most people as a dingy, dirty, smutty, smoky old place to work in with poorly lighted and ventilated buildings. No wonder in many cases the best boys do not take as kindly to the trade as to some other calling. It becomes necessary then for us to give all these matters more attention, to devise ways of bettering the conditions of the very places in which we expect our men to perform their daily task. I do not wish to be understood as advocating a rank waste of money in providing expensive buildings, equipped with lavatories, with the latest design of shower baths, individual wash bowls with soap cake, Turkish towels, elaborate dining rooms, etc., but I do wish to say that in my judgment it is possible for any foundryman to provide a suitable place for his men to hang their coats and deposit their dinner pails upon entering the shop; that such a place be so arranged as to insure neatness and cleanliness, as well as a suitable place for the workmen to warm their coffee at noon, and a small, well-kept room for them to gather in in cold weather and eat their noon meal in comfort. As an attraction for the best quality of boys, it will be found that a well-lighted shop beats darkness; that a neat and well-appointed locker is superior to the peg in any old post for the coat; that a comfortable place for dinner is ahead of the sand pit; that conveniently-located, well-kept closets surpass the back yard, and that a suitable place to wash up at the shop is more attractive to high grade boys than carrying home the day's accumulation of sand and dirt to perfume the home and vex the family.—*O. P. Briggs in American Industries.*

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MACHINERY

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Attention is called to the paragraph headed "Caution" on page 107 of the Engineering Edition, and page 67 of the Shop Edition.

* * *

A subject upon which more data could profitably be published is that of the cost of machine work. The article recently published in our columns upon "Estimating the Cost of Machine Work," by J. A. Webster, has attracted considerable attention, but it deals with only one machine, and that a very small one. We should like, for example, data upon different classes of work, with an explanation of the methods followed in estimating the cost of such work. Take the case of the screw-machine. How do you estimate the time for screw-machine work? What allowance do you make for setting up, for the actual cutting of the stock, for the rotation of the turret, etc.? To what make of machine do these figures apply? Can you give data upon actual pieces that have been made? The same kind of information could also be given in regard to milling machine, or lathe, or planer work, and the subject could even be carried so far as to cover the cost of equipping a plant for turning out certain classes of machinery; or of making jigs and fixtures for a certain piece; or of equipping a tool room for a manufacturing plant of a certain size and kind. These suggestions, we think, will be sufficient, and we trust that some of our readers will find it convenient to contribute notes along the lines indicated.

* * *

One would infer from a recent item in the daily press, that the Association of Licensed Automobile Manufacturers, who control the famous Seldon patent, had scored an important point through an injunction in the case of one Moore, a man who recently came to this country, and attempted to operate an imported gasoline car. But it is not so. The Seldon patent is supposed to be a basic patent upon the combination of a gasoline engine, a vehicle and clutch connecting the engine with the driving axle of the vehicle. This patent is the tie that binds together the Association of Licensed Automobile Manufacturers and this organization is attempting to prevent any but licensed cars being either manufactured or used. The *Horseless Age* explains that the injunction against Moore has no particular bearing upon the main point at issue. When the validity of a patent is once sustained, the owners of the patent can have injunctions issued against all infringers, but no court ruling has yet been rendered with respect to the validity of the Seldon patent. The injunction against Moore was not issued for infringing the patent, but because he failed to put in a defense and so lost the case by default. Litigation is now in progress between the Ford Motor Company and the Association of Licensed Automobile Manufacturers, and this will probably be the first decision with respect to the validity of the patent.

* * *

WHY THE TURBINE HAS SUCCEEDED.

One of our contemporaries has been explaining why the steam turbine has met with phenomenal success, and has risen above the horizon of the steam-power field with startling rapidity. It is pointed out, that while we have heard about the turbine for only a few years, inventors have really been at work upon it for many years; that as the steam engine has now reached its highest state of development, it is in order for something else to come to the front; that as the steam turbine has proven to be efficient, simple and durable, it must, as a matter of course, displace the reciprocating engine for many purposes; and so on, outlining from beginning to end the history and advantages of this prime mover.

It is safe to say that the turbine could not have automatically displaced the steam engine to the extent that it has. Such displacement is due to the fact that there were hard pushers behind, granting, of course, that the turbine has merit.

A few years ago the two large electrical companies of America found it desirable to manufacture their own prime movers, as well as generators, for the large power plants they were so frequently called upon to equip. The indications were that power would thereafter be concentrated more and more in large central stations. Big power units require big firms to build them, and above all, to market them, and the question was whether these two big concerns should build steam engines or steam turbines. One of them did, in fact, start to build large vertical engines; but the turbine had such possibilities that both eventually decided in its favor, and their immense resources were directed toward perfecting the steam turbine. They have succeeded in bringing the turbine to a high state of efficiency in a remarkably short space of time; and what is more to the point, in scooping the orders for large generating units. It is not to be supposed for an instant that the turbine could have reached its present status had it not been backed by the resources of large and wealthy companies, having extensive business connections. The development of the Parsons turbine in England extended over many years, previous to its introduction in this country, and if the Curtis turbine, for example, had been taken up in a small way, as was the Parsons turbine at first, it would also have required many years to have given it its present standing in competition with the products of the leading engine builders. An analysis of the situation on the continent will show that practically the same thing is transpiring. It is the large firms, or combinations of such firms, that are succeeding with the turbine. The small firm must work more slowly and carefully, but we believe that many of the "little fellows" will eventually figure in the turbine industry, and be successful competitors.

* * *

There is a movement under foot to establish a school to train textile salesmen, following the plan of the Lowell Textile School, where students learn to design textile fabrics and to carry through the processes of manufacture from the spinning room to the inspection bench. It is contended that those who sell textile products ought to know something about their manufacture, as well as to be instructed in the devious ways of the wily salesman. Those who have occasion to visit the stores in our various cities where machinery is sold may well wish that something of the kind might be done for the innocent salesman who attempts to explain the intricacies, or in many cases, even the simplicities, of the modern machine tool. Complex or simple, however, the salesman is sure to become tangled up and to flounder hopelessly when he gets beyond the "superior workmanship" or "highest quality" stage of the discussion and tries to enter the field of mechanics.

ENGINEERING REVIEW.

CURRENT EVENTS, TECHNICAL AND MECHANICAL—LEADING ARTICLES OF THE MONTH.

Although the steam turbine of the Parson type is an exceedingly simple machine in the matter of operation, there is one feature of the construction which is not so simple, and that is the setting of the multiplicity of blades. This work not only requires care and skill, but it is an enormous task on the larger sizes. For example the rotor of one of the turbines of the steamer *Victorian* contains 750,000 blades according to a statement in the September issue of *Marine Engineering*. This vessel, by the way, is the first turbine steamer built for Trans-Atlantic service. It will ply between Liverpool and Montreal. It is 540 feet long, 60 feet broad and 42½ feet deep, having a capacity of 10,630 tons and accommodations for 1,500 passengers.

A recent consular report says that about twelve hundred flint-lock muskets are turned out weekly in Birmingham, England, and that a large number of this antiquated firearm is also made at Leeds, Belgium. It appears that these guns are sent to Central and East Africa for use by the natives, to whom the possession of modern rifles is denied by law; but this market is not the only one enjoyed by the manufacturers, it being quite the fad for American travelers to buy such weapons under the impression that they are getting something quite antique. It is said that the demand for old weapons is so great during the American tourists' season that the genuine article would go but a short way to supply it.

A correspondent of *Engineering* (London) says that in designing some tables for a slide rule, he tried to devise a simple formula for obtaining the sine of an angle when no printed tables were at hand and worked out the following formula which gives a very close approximation below 80 degrees:

$$\text{Radius} = 1000.$$

$$\frac{\theta = a.}{10}$$

Then $$\sin\theta = 174a - (a^3 \times 0.81).$$

Examples.

Angles, deg.	Sine by Tables.	Sine by Formula.	Error.	Error.
5	87.1	86.8	0.3	1 in 286
20	342.0	341.5	0.5	1 in 684
40	642.8	644.2	1.4	1 in 460
60	866.0	869.1	3.1	1 in 280
80	984.8	977.3	7.5	1 in 131

Beyond 80 degrees the error increases rapidly. For all practical purposes, the coefficient 0.81 may be taken as 0.8.

The superheated steam problem has been so intensified by the advent of the steam turbine that some definite data on the subject may be expected before long. The reports from one class of users, whose power plants carry a steady load, are most favorable; they are having no trouble to speak of and are satisfied with the results. There is another class, however, whose remarks on superheat are wholly unfit for publication. They have been trying to use high superheat with power units carrying loads subject to sudden and heavy fluctuations. The result of these is an occasional excessive rise in temperature, which causes all sorts of trouble in the piping and valves. It has even caused havoc in steam turbines, if numerous current reports have any substantial basis of truth. This is not at all surprising, for the clearances of turbines are very small, and the rapid and high rise in the temperature of superheated steam when a heavy load is largely thrown off with suddenness might reasonably be expected to cause overheating of the rotors and guides, with attendant binding. It is understood that the regulation of superheating is now receiving the attention of a number of specialists.—*Engineering Record*.

Assuming that coal would be used in the plant with a value of 12,000 B. T. U. per pound, and that the efficiency of the boiler be 60 per cent., each pound of coal will transmit to the boiler 7,200 B. T. U. Since each pound of water takes up 30 B. T. U. on its passage through the heating boiler, 1 pound of coal will heat 240 pounds or 28.8 gallons of water. This is equivalent to supplying under extreme conditions of heat loss, 28.8 square feet of radiation. In condensing, low-pressure steam gives up approximately its latent heat, or about 966 B. T. U. per pound, and since each boiler horse-power is equivalent to 34.5 pounds of water evaporated at atmospheric pressure, we have 1 boiler horse-power equal to 33,327 B. T. U. Now since each pound of coal transfers to the water 7,200 B. T. U., 1 boiler horse-power would require $33{,}327 \div 7{,}200 = 4.63$ pounds of coal. One pound of coal we found to supply 28.3 square feet of radiation, consequently 1 boiler horse-power would, from the above figures, supply $4.63 \times 28.8 = 133.24$ square feet of radiation. Boilers are usually estimated for service per each 100 H. P., consequently a 100-H. P. boiler would supply 13,324 square feet of water-heating surface.—*Domestic Engineering*.

That the cast-iron car-wheel can be made stronger and more durable by making it more elastic is an idea advocated by the *Railway Master Mechanic*. It holds that the popular form of the double plate wheel must be abandoned, for that type is most rigid of all and contains metal where it is least needed. By reducing the rigidity of that portion between the rim and the hub the wheel will be more flexible, both laterally and vertically, thus making it better able to withstand shocks. With the more flexible construction the flange, which is now the weakest part, will in effect be strengthened, as the shock will not be localized on the root of the flange, but will be transferred more largely to the body of the wheel. The flange is the only part of the modern car-wheel that has not been enlarged, simply because it cannot be thickened on account of the limits imposed by frogs and crossings; but the need of a thicker flange, it is believed, will disappear with the advent of the wheel that has better material and with the material so placed as to combine strength with minimum rigidity. A number of 700-pound wheels which closely conform to the ideas expressed above are now running under 100,000-pound cars successfully. The wheels are made of charcoal iron and this seems to be the metal that will necessarily be used in improving the cast-iron wheel.

One of the papers presented before the National Electric Light Association at the June convention held in Denver and Colorado Springs, Colo., consisted of kinks and wrinkles contributed by stationary engineers in various parts of the country. One of these wrinkles, contributed by Mr. G. H. Cushman, San Antonio, Texas, recommends the use of chain drive to replace the time-honored governor belt as follows:

"Since it is the duty of a governor on an engine to adjust the supply of steam admitted to the cylinder to suit the load, the medium of transmission of power to drive the governor must act as quickly as possible. It is the duty of the fly-wheel to take care of the variations of the load instantaneously, while governor adjusts same by controlling the steam—taking a comparatively longer period of time to do so. We also know that under best conditions possible there is belt slippage and furthermore that a belt may break without giving any warning. To overcome these deficiences we have replaced the governor belt on three engines by a silent chain-drive, and we are well pleased with the change. To do this a special sprocket wheel had to be made to fit over the engine shaft; a sprocket was also made to replace the governor pulley. The silent chain-drive has four advantages over the belt, in that it cannot slip, it does not stretch, oil does not ruin it, and it is stronger."

In a recent issue of the *Patternmaker*, Mr. Joseph L. Gobeille, of the Gobeille Pattern Co., Cleveland, Ohio, tells how he estimates the weight of ornamental light castings from the patterns. The plan will be recognized as a very old idea, originating, we are told, with that ancient worthy Archimedes, but its application to wooden patterns has the savor of novelty:

"In estimating the weight of a cook stove from the wood

patterns, the ordinary practice is to weigh the patterns, allowing a pound of iron for each ounce indicated, and then adding thereto ten per cent. This method is fairly accurate with patterns made entirely of wood; but take an oven door after it is ornamented and there are other materials to account for, such as composition, varnish, and very often lead if the ornamentation be heavy and bold. Estimating weight by this formula on such a pattern is like looking at the sun to tell the time of day. You will make a close guess, perhaps, but it will be only a guess. To get accurately at the solution of such problems, I made a tank 24 × 36 inches and 12 inches deep, and for 3 inches of the top I graduated a scale into 1-64ths inches. This tank was leveled up and filled with water to the lower lines of the scale. The pattern, well-varnished, was plunged into the tank for a moment, the displacement noted, then taken out and wiped off. If the reading of the scale showed a rise of 1/8 inch, then I knew that the casting, under ordinary circumstances, would weigh thirty-one pounds when set up in the finished stove. The constancy and accuracy of this method are marvelous and have been a source of wonder to many foundrymen to whom I showed my simple expedient for the first time."

POWER SAVING IN MACHINE SHOPS.

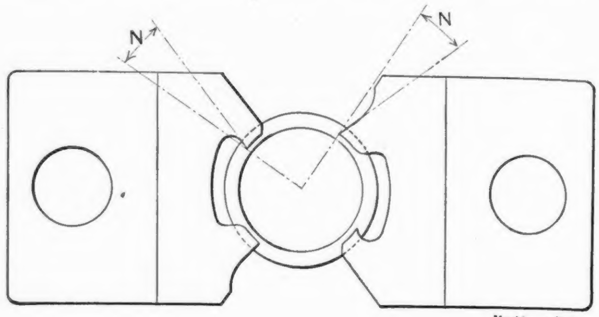
Electricity, August 23, 1905.

Machine shops in the United States are predisposed to improvements. They are the most progressive of all our institutions. They are places where a man's time is valuable enough to call for as much care in its utilization as circumstances will permit. Every hour of time saved by the intelligent manner in which work is done not only means profit to the shop but it means as well a reputation for dispatch. For this reason the machine shop has become the scene of the use of automatic appliances. It has become a true engineering center in which is focused the inventiveness, skill and labor of the best class of our constructors. And in addition, it has led to the investigation of certain important problems in shop economy which relate to the cost of turning out work, and in some respects, to the time as well. The features of this kind, to which attention is directed, are those of shaft and belt friction, and the cost of this manner of transmitting and distributing power, in comparison with the more up-to-date method of using individual motors for each machine. Tests of the efficiency in the first case gave rise to an interesting series of results, among the most important of which was a surprisingly high figure for the wasted power. This, as may be imagined, was only due to the friction developed by the stiff belts and countershafts employed in these shops, and it is not difficult to trace this waste again to the thickening of oil in the bearings and the change in the flexibility of the belting. In a shop using 1,000 H. P., the countershafting consumes nearly 400 H. P. From a dollar and cents standpoint, estimating the cost of a horse power at $50 a year, the extent of this loss can be measured up in the annual waste of $20,000. An installation of motors would naturally represent a greater expenditure in the first place, but the efficiency would readily average over 80 per cent net and perhaps 90. From this standpoint, a saving of over $10,000 appears. A saving of this kind certainly warrants an investment of at least five times as much, or $50,000, in electrical transmission machinery. Machine shop saving in this respect, in large shops at least, means a profit in the operating expense. Modern machine shop practice is a scientific as well as a financial and mechanical proposition.

SHAPE OF DIES FOR THREADING STEEL PIPE.

Railroad Gazette, July 21, 1905.

Since the introduction of modern weldable Bessemer steel for pipe considerable trouble has been experienced in cutting satisfactory threads on it. The National Tube Company, Pittsburg, Pa., investigated the matter with a view of finding the cause of the above trouble. After much experimenting the die department of the company found that the trouble was largely due to insufficient rake and clearance being given to the threading of dies. Without the proper rake and clearance the thread is ragged and torn and the tool wears quickly. There are also other points peculiar to the construction of threading dies which have to do with the results obtained, but all things being considered it was found that the angle of rake (*N* in the sketch) is the point which has most to do with efficiency, and is also the feature which is generally overlooked in the tools commonly supplied to the trade. A properly designed die is particularly necessary for threading soft steel pipe, but at the same time a correctly shaped die will work to better advantage on iron pipe than one improperly made, as truer threads and consequently tighter joints will be the result as



Machinery,N.Y.

well as lower cost of threading and less damage to the pipe and dies. A die, to work properly and to give a clean, true thread, must have the proper amount of rake, as shown in the illustration; the chasers must be rigidly held and controlled close to the work with a due regard to chip room, and sufficient lubricating oil must also be employed. The following angles between a line through the center of the pipe and the cutting face of the chaser at the point of contact with the pipe are submitted by Frank N. Speller, metallurgical engineer of the National Tube Company, as giving the best results in practice:

Size.	Angle N.	Size.	Angle N.
1-inch........	26 degrees.	6-inch........	22 degrees.
2-inch........	25 "	8-inch........	20 "
3-inch........	25 "	10-inch........	20 "
4-inch........	23 "	12-inch........	18 "

TIDAL POWER.

Practical Engineer, August 11, 1905.

The idea of tidal motors has long been an attractive one, and as an idea there is much to be said for it. There is no perpetual motion madness nor any fundamental or mechanical unsoundness in a proposal to utilize for power purposes the rise and fall of water brought about by tidal influences. Any engineer could undertake to get work done by the tide. But he could also do the work very much cheaper, and with greater convenience to all concerned, in other ways. An engineer has been defined as a man who can do for one dollar that which any fool could do for two. And such a definition has obviously its limitations, but it will serve to remind us that in tidal power schemes, as in most other things, we have to consider not merely the cost of carrying them into effect, but such cost as compared with the cost of attaining the same result another way.

Of the many proposals put forth in connection with the harnessing of the tide for industrial purposes, the simplest and probably the oldest comprises the employment of a pontoon or large float, which is raised by an incoming and falls with an outgoing tide. Various mechanical contrivances, some of considerable ingenuity, have been devised for the conversion of the slow reciprocating movement of the float into a rotary motion in one direction of a shaft, from which the power is taken by any suitable transmitting gear. From a mechanical point of view many of such arrangements leave little if anything to be desired. Their non-success arises from the fact that for obtaining an appreciable amount of power a float of very great size is required. As an example, let it be assumed that the float be placed at a position where there is a mean difference of as much as 36 feet between high and low water. Taking it as six hours between high and low water, the float will descend through a distance of 36 feet, which represents an average of 6 feet an hour or one-tenth of a foot per minute. If, therefore, we require to obtain only 10 H. P. throughout that period, it will be seen that the float

must have a weight of no less than 33,000 × 10 ÷ 1-10 = 3,300,000 = nearly 1,500 tons.

The foregoing very simple figures will give some idea as to the size of pontoons required for the supply of the power necessary for, say, a small electric-lighting station. As we write there occurs to us the remark made on such a proposal by a well-known engineer, to the effect that according to his calculations a pontoon as big as a house would only give one cat power.

The practical uselessness of the float or pontoon for tidal motor service has led, no doubt, to the suggestion of impounding reservoirs, and the employment therewith of water wheels or turbines. Many forms of such an arrangement have been set out on paper, and some of them look quite attractive and of considerable promise. But here again the difficulty of size is met with. Very large reservoirs and the employment of much plant will yield but a small quantity of power. There are, we believe, some instances in this country where, on a small scale, something of this kind has been accomplished with advantage, but in general it does not appear that there are many places where the utilization of tidal waters would be worth attempting. The thing can be done, but not with profit.

CASTING BRASS OR BRONZE ON IRON.

Practical Engineer, August 11, 1905.

For some purposes it is desirable to cast brass or bronze on to iron, and in such cases there is usually trouble, as the alloy comes more or less spongy. Nevertheless, such work can be done, and done well if the thickness of the alloy is sufficient, and if the following hints are followed satisfactory results will be obtained after a few trials. Say that we take

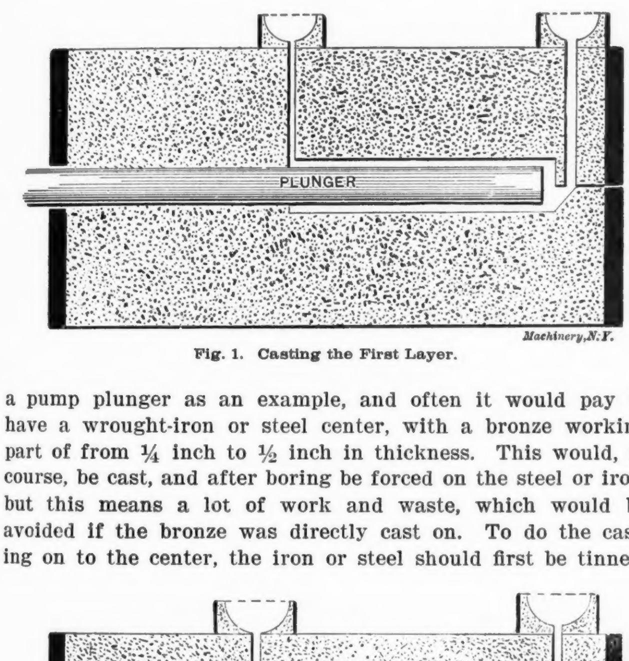


Fig. 1. Casting the First Layer.

a pump plunger as an example, and often it would pay to have a wrought-iron or steel center, with a bronze working part of from ¼ inch to ½ inch in thickness. This would, of course, be cast, and after boring be forced on the steel or iron, but this means a lot of work and waste, which would be avoided if the bronze was directly cast on. To do the casting on to the center, the iron or steel should first be tinned,

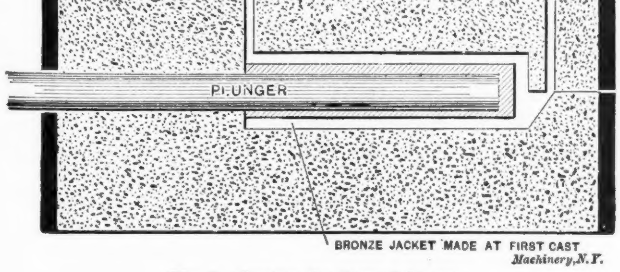


Fig. 2. Casting the Second Layer.

if facilities are at hand, but if this is not a conveniently performed operation it may be neglected, simply making the iron or steel quite clean and bright.

When all is ready, a mold should be made to give half the thickness of bronze in the center, and to a rather greater length than the finished covering is to be, the center being then put in as a core, and fluid metal poured to fill the mould, as shown in Fig. 1, care being taken that both heads are full. There will be some spluttering if the core is untinned, but by pouring at the end this will not cause much trouble, although a few splashes may fly about. A long box will have to be used, and in many cases one end of both drag and cope will have to be cut away to permit the steel or iron plunger to pass through; with short ones, however, this will not be necessary.

When practically cold, the work should be taken from the mold, the runner and riser cut off close, and the bronze thoroughly cleaned from sand and brushed up bright, having it ready for placing in the second mold.

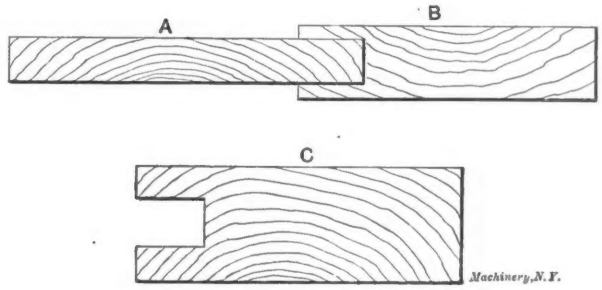


Fig. 3. Pattern with Interchangeable Bodes.

The second mold will be prepared in the same way as the first one, but, in addition to an allowance for shrinkage, an allowance for turning must be made, and when the plunger is in position a section of the mold would appear as in Fig. 2. In this case the metal will be run through the mold, a little overflowing from the riser into a pool on top of the box, as this removes any dirt that may have risen to the top of the mold, and also insures the soundness of the bronze cast on. When cold the bronze is turned down in the ordinary way, and should be solid. The porousness of the first metal cast on the iron or steel affords a good keying for the second lot cast, while the contraction of the metal in cooling holds it immovably on to the iron or steel. Either brass, bronze, or phosphor bronze can be cast on in the way described, provided it is run in a fluid state, and well skimmed before pouring. Preferentially, the pouring should be done with a good head of metal, and the entrance to the mold should be in the drag, as shown in the illustrations.

The patterns for a plunger should be in three pieces, as shown in Fig. 3, and may be well made in pine. *A* is the "print" for the plunger, and is turned up exactly to size; *B* is the pattern for the first cast, and *C* that for the second cast, the part *A* being made to slip firmly into either *B* or *C*, as occasion requires. Other patterns will be made on the same principle for double casting, and will, of course, vary with the articles to be treated.

PREPARING DEAD SURFACES ON BRASS.

Brass World.

The dead dip is the one which is used to impart a satiny or crystalline finish to the surfaces of ornamental brass articles. The bright dip gives the smooth, shining, and perfectly even surface, but the dead dip produces a surface which is most beautiful. When properly done there is just enough life to it to give a pleasing appearance, but yet not sufficient to give false light reflections like a highly-polished surface. It is by far the most pleasing of any dip finishes, and can be used as a base for many secondary finishes. Sulphate of zinc in a finely-divided form is necessary for the dead surface, and is the essential difference between it and the bright dip. The most modern method of making up the dead dip is to produce the sulphate of zinc directly in the solution and in the precipitated form.

Take 1 gallon of yellow aqua-fortis (38 deg.) and place in a stone crock which is surrounded with cold water. The cold water is to keep the heat, which is formed by the reaction, from evaporating the acid. Now add metallic zinc in small pieces at a time until the acid will dissolve no more. The zinc may be in any convenient form: sheet clippings, lumps,

granulated, or any other shape which is such that it may be added little by little. If all is added at once, the action is so violent that it will boil over. When the acid will dissolve no more zinc, it will be found that some of the acid has evaporated by the heat, and it will be necessary to add enough fresh acid to make up the original gallon. When this is done add 1 gallon of strong oil of vitriol. The mixture should be stirred with a wooden paddle while the oil of vitriol is being added.

As the sulphuric acid is being added it will be noticed that the solution begins to grow milky, and finally the whole has the consistency of thick cream. This is caused by the sulphuric acid (oil of vitriol) precipitating the sulphate of zinc. Thus the very finely-divided precipitate of sulphate of zinc is formed. If one desires to use known quantities of acid and zinc, the following amounts may be taken:

Oil of vitriol	1 gallon
Aqua-fortis (38 deg.)	1 gallon
Metallic zinc	6 ounces

In dissolving the zinc in the aqua-fortis it is necessary to be sure that none remains undissolved in the bottom, as this would spoil the results.

The dead or matt dip is used hot, and therefore, is kept in a stone crock surrounded with hot water. To use it, the articles to be matted are polished and cleaned in the usual manner, and the dip thoroughly stirred with a wooden paddle so as to bring up the sulphate of zinc which has settled to the bottom. Now dip the work in the solution and allow it to remain until the requisite matt is obtained. This is a point which can be learned only by experience. When the brass article is first introduced, there is a rapid action on the surface, but in a few seconds this slows down so that there is scarcely any. Now remove the article, and rinse and immediately dip into the usual bright dip. This is necessary for the reason that the dead dip produces a dark coating upon the surface, which, were it left on, would not show the real effect or the color of the metal. The bright dip, however, removes this, and exposes the true dead surface.

The usual rule for making up the dead dip is to use equal parts of oil of vitriol and aqua-fortis, but these may be altered to suit the case. More oil of vitriol gives a finer matt, while a larger quantity of aqua-fortis will give a coarser matt. When the dip becomes old it is unnecessary to add more zinc, as it never requires it, on account of a little going into the solution each time anything is dipped. After a while, however, the solution becomes loaded with copper salts, and should then be thrown away.

A new dip does not work well, and will not give good results when used at once. It is usual to allow it to remain over night, when it will be found to be in a better working condition in the morning. A new dip will frequently refuse to work, and the addition of a little water will often start it. The water must be used sparingly, however, and only when necessary. Water, as a usual thing, spoils a dead dip, and must be avoided. After a little while it may be necessary to add a little more aqua-fortis, and this may be introduced as desired. Much care is needed in working the dead dip, and it is something that requires constant watching and experience if uniform results are to be obtained. The chief difficulty in working the dead dip is to have to match a given article which is brought in to the dipper. No difficulty is found in producing the dead dip, if the solution is made up properly, but to have to match what is on a sample that has been submitted is one that tests the skill of a dipper more than anything else. The only way that it can be done is to "cut and try" and add aqua-fortis or oil of vitriol as the case requires. The dead or matt dip can be obtained only upon brass or German silver. In other words, only on alloys which contain zinc. The best results are obtained upon yellow brass high in zinc.

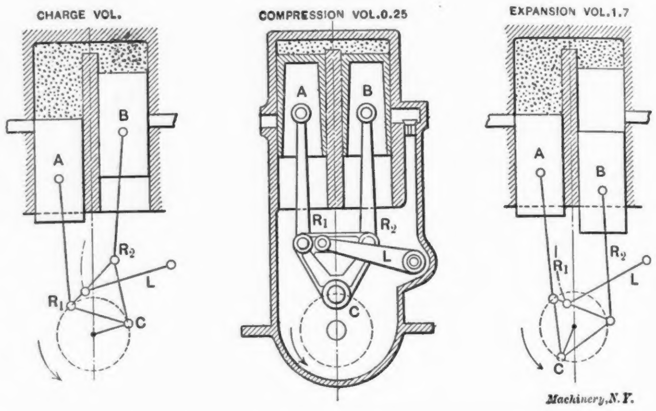
THE LISTER TWO-CYCLE ENGINE.

The Engineer, September 15, 1905.

Two parallel cylinders, A and B, are provided, the latter being the inlet cylinder and the former the exhaust, a compression chamber, which is common to both, being arranged between the two cylinders. Each cylinder has its piston connected to a triangular frame C by rods denoted R_1, R_2. One angle is coupled to the crank pin and the movement of the frame is controlled by the radius rod L, which is pivoted to the casing of the engine.

As indicated in the diagrams, the charge is expanded to 1.7 its initial volume before exhaust takes place, and the combined effective power strokes of the two pistons are said to be approximately equal to 1.8 the crank stroke, the compression portion of the return stroke being equal to 1.2 the crank stroke. The designer, Mr. Lister, of Keighley, Eng., holds that an engine of this type can be constructed with a small unit of weight per horse power and a high efficiency, at the same time occupying but small space.

When the crank is rotating in the direction of the arrow, as noted in the middle view, commencing with the position at the completion of the compression stroke, the pressure which is generated acts simultaneously on both pistons as the



ignition takes place. Piston A follows piston B in its travel, and the latter has approached the end of its out stroke when piston A reaches the exhaust port. The expansion figure shows the volume at this point, the exhaust then taking place and the pressure immediately drops to that of the atmosphere. An inlet port provided with a back pressure valve connected by a passage to the enclosed crank chambers, then having been uncovered by piston B, allows a volume of air under pressure to pass into the cylinder B as a scavenging charge sweeping out the waste gases from the explosion preceding, and thoroughly cleaning the compression space and cylinder of those burnt gases.

Again, in advance of piston A, piston B commences its up stroke, and forces out a further volume of contents from the cylinder A, and approaches the end of its stroke at the time when the piston A is closing the exhaust port. The volume before compression is then as shown in the charge diagram at the left. A charge of oil or other liquid fuel is pumped into cylinder B shortly before the closing of the exhaust port, in such a way as to mix intimately with the air in the cylinder, forming an explosive mixture which is ignited after being compressed.

This cycle is repeated at each revolution, so that this engine, in common with all two-cycle engines, has an operating impulse at every revolution of the single crank shaft, thus differing from the engines of the four-cycle or Otto principle, which have an impulse every two revolutions.

Mr. Lister claims that the volume of cylinder contents at the point of exhaust is from 50 to 70 per cent greater than the initial volume of the charge before the commencement of compression, resulting in a noiseless exhaust, better combustion, and increased efficiency. He holds that with the free exhaust port the loss from back pressure is avoided, and with this design of engine there is a positive scavenging action without any loss of incoming explosive charge. Another feature of utility claimed for this engine is the rapid expansion with consequent reduction in the cylinder wall losses. There is also a definite full charge of explosion mixture at all speeds and under all conditions of temperature. The short connecting rods have a greatly reduced angle of pressure which does not exceed 5 degrees at any point of the working stroke. This is of great importance as the lubrication is excellent and the cylinder and piston wear are equally distributed.

MACHINE FOR ROUNDING AUTOMOBILE TRANSMISSION GEAR TEETH.

The *Revue De Mecanique* of Paris, publishes a description of a machine for rounding the ends of the teeth of the gears used in the sliding gear transmission system for automobiles, which ought to be of interest to American manufacturers. In Fig. 2, *L* is the head carrying the work arbor on which may be seen mounted the work. This arbor is revolved continuously by gear *D*, through change gears, and a nest of gears in box *E*. The cutter is mounted in the end of the

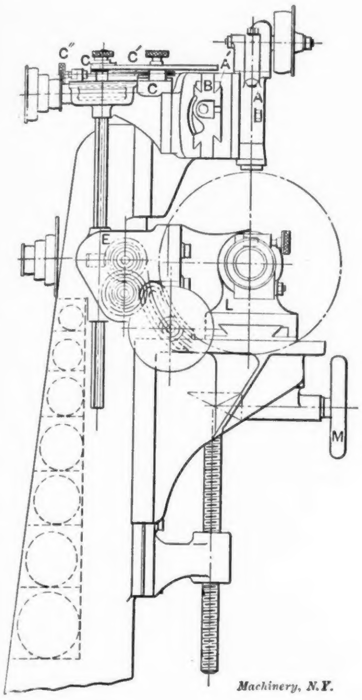


Fig. 1.

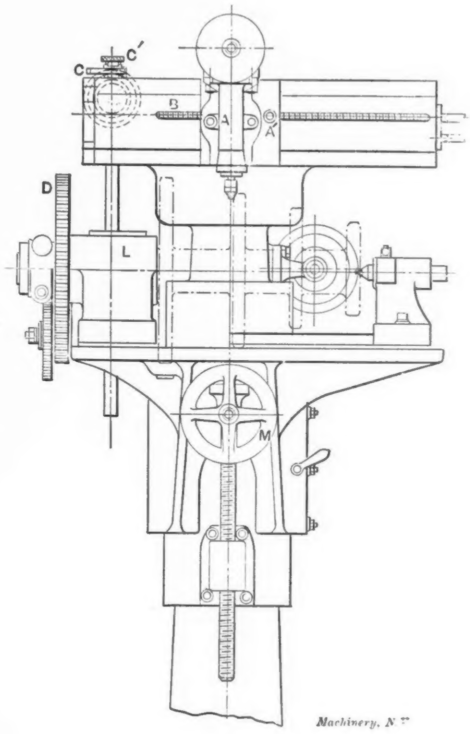


Fig. 2.

spindle in head *A*, which has a slight swiveling adjustment. It is driven through bevel gears by the pulley seen at the top of the head. This head *A* is fixed to the slide *A'*, which may be clamped at any point on the crossrail *B*; the slide is given a reciprocating motion by means of lever *C'* seen in Fig. 3 and cam *C*. Cam *C* is driven through a worm and worm wheel by the cone pulley plainly shown in the top view. The shaft on which this cam is mounted extends down through to the change gear box which governs the movement of the dividing wheel *D*.

In the operation of the machine, with the spindle *A* revolving continuously, and the work being rotated through the change gears at the proper rate, the cam will move the cutter in and out of the spaces between the teeth, as they present themselves with such a movement that the cutter will give the desired profile to the corner of the tooth. The cone of gears in gear case *E* may be given such ratios that they will correspond to the ratios of the transmission gears whose teeth are being rounded. In this case, in changing from one gear to another in the series, the push pin of the nest of gears will simply be pushed in or out to correspond with the gear being cut. The lever *C'* may be located on either side of the cam *C* so that both sides of the gear may be operated on without its being necessary to reverse the work in the machine. A spring is concealed in the crossrail *B* which opposes the movement given it by the cam; the pressure imposed by this spring may also be reversed by turning the shaft which projects at the end of the crossrail. In working on gears of varying diameters the work-table is elevated or depressed by hand wheel *M*. For varying the outline of the round which is given to the corner of the gear tooth, the pivot of lever *C'* is moved in or out by the knurled head screw *c''*. It is unnecessary therefore to change the cam except where a radically different shaped tooth corner is required, such as a bevelled corner, for instance.

This machine is built by M. Marcel Lejeune, 93 Rue d' Angolême, Paris, France.

OIL FURNACES.*

When the larger part of our output was the result of hand labor, the heating of the material was a simple question, the quality of heat being the only consideration necessary, as any of the coal or coke fires could easily supply sufficient working material, but at present a very large percentage of our work is machine blacksmithing and it is absolutely necessary to supply the various machines with raw material, properly heated, and in large enough quantities to work them to their full capacities. These machines cost the railroad a large amount of money and if not turning out finished material in quantities approximating their capacities, are expensive investments.

In looking for a method of increasing heating facilities for machine work, petroleum, so bountifully supplied by nature, was naturally a great attraction. Crude petroleum or fuel oil, which is commonly used, has a very high heating efficiency containing from 20,000 to 22,000 heat units per pound as compared with from 12,000 to 14,000 per pound of coal.

It is capable of practically perfect combustion, leaving no ash, and when properly handled producing no smoke. The quality of the heat is as satisfactory as coal or coke, if generated in the proper manner.

The mere act of burning oil is a simple one, but the burning of oil and producing a proper flame is a very particular process. Atomizing the oil with air or steam under pressure through variously constructed burners and burning the resulting mixture by depending upon the heat of the furnace to keep up the combustion, is a method commonly followed. This process necessitates forcing unconsumed oil, no matter how thoroughly atomized, into the furnace. Combustion does not begin until this part of the operation has been performed, and the atomized oil has been spread over the heating area and surfaces of the furnace. More or less of it, before burning, comes in contact with the material being heated, where it carbonizes, causing imperfect combustion, and a cooling effect upon the material, thus necessitating a longer period to properly heat the iron. Again, this high pressure or atomizer system almost invariably produces an oxidizing flame, resulting in the burning of the material; bridge walls and other arrangements, somewhat help the combustion but do not overcome the oxidizing

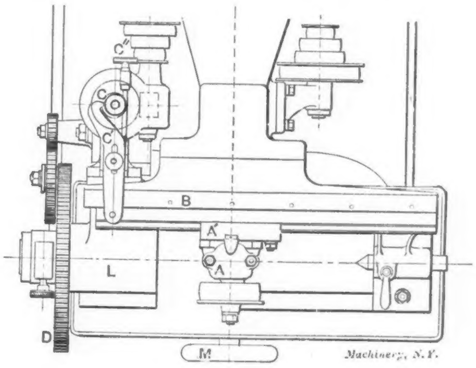


Fig. 3.

flame. Owing to our familiarity with compressed air due to its general use in our shops, few of us appreciate the high cost of producing it and its influence in increasing the cost of heating where used as an atomizing agent in oil furnaces.

These several objections to the atomizer system were appreciated back in the early 90's by a brother railroad man, and to his thorough study of the subject and his years of experimental work we are indebted for ideas which have been developed into the low pressure and economical process known by his name to-day. He eliminated the expensive

* Paper by Mr. John McNally, of Chicago and Northwestern Railroad.

compressed air and substituted the ordinary low pressure fan blast used in our open forges, on the theory that volume and not pressure was necessary for the successful burning of oil. With a low pressure of only a few ounces he fortunately could not successfully atomize, so was obliged to work on other lines, resulting in a process which first burned the oil in a small amount of air, thereby breaking it up into the hydro-carbons, then with heat of the subsequent combustion breaking these hydro-carbons down to gases, such as marsh gas, carbon monoxide, free hydrogen, etc., and finally supplying these gases with a second and larger volume of air burning them to completion. All of this being done in a small and independent combustion chamber, the hot gases only being forced into the heating area of the furnace. This process resulted in a very perfect combustion of the oil at a low cost of operation and supplied a soft, dry heat to the furnaces.

Very thorough and extensive tests made on this process by Prof. Kenosche at Iowa State College during the past winter, demonstrated a combustion practically perfect before entering the heating area of the furnace, a temperature above 3,000 degrees F. at the same point, and the important fact that practically no oxygen was unconsumed, resulting in a reducing flame which could not injure the material being heated.

Other low-pressure processes discharging a mixture of oil and air into a combustion chamber or furnace direct are an improvement on the high-pressure system, but do not as completely burn the oil, owing to the difficulty of supplying a sufficient quantity of oxygen at a single point, and the result is a damp oxidizing flame.

The process we are using at the Northwestern shops overcomes this difficulty by means of the secondary air supply. Having perfected the combustion, and thus insured a maximum amount of heat from the least bulk of oil, the furnace design and construction to save and use this heat, is equally important. The percentage of loss, due to radiation, is often high, but with proper construction of furnace can be made a small item. With perfect combustion stacks are not necessary and a great loss of heat can be prevented by not using them. Radiation which makes uncomfortable conditions for the men is another source of loss and should be overcome, and when well done will greatly increase the efficiency.

A pound of oil contains a given amount of heat, all of which is made available with perfect combustion, but a poorly designed and constructed furnace, even if it is apparently doing satisfactory work, may be losing enough of this heat to make the cost of the operation prohibitive. On the other hand comparative costs of coal and coke fuels depend upon locality; with properly constructed and operated oil furnaces the fuel bill can be much higher than coal or coke would be in the same furnace and still the cost of the work produced be materially lower. This is due to the greatly increased output, the greater intensity of heat, the elimination of tending fires, the short time required to bring an oil furnace to the desired working temperature, and the improved conditions under which the furnace men work.

1. The maintaining of an even heat of any desired temperature is assured with a properly constructed oil furnace.
2. The quality of work is of a higher grade.
3. The quantity of material turned out is greatly increased.

* * *

TWO GERMAN LATHE ATTACHMENTS.

The *Zeitschrift fur Werkzeugmaschinen und Werkzeuge*, in a recent review of various machine methods, illustrated the lathe attachments shown in the accompanying drawings, for turning concave and convex spherical surfaces. The first one,

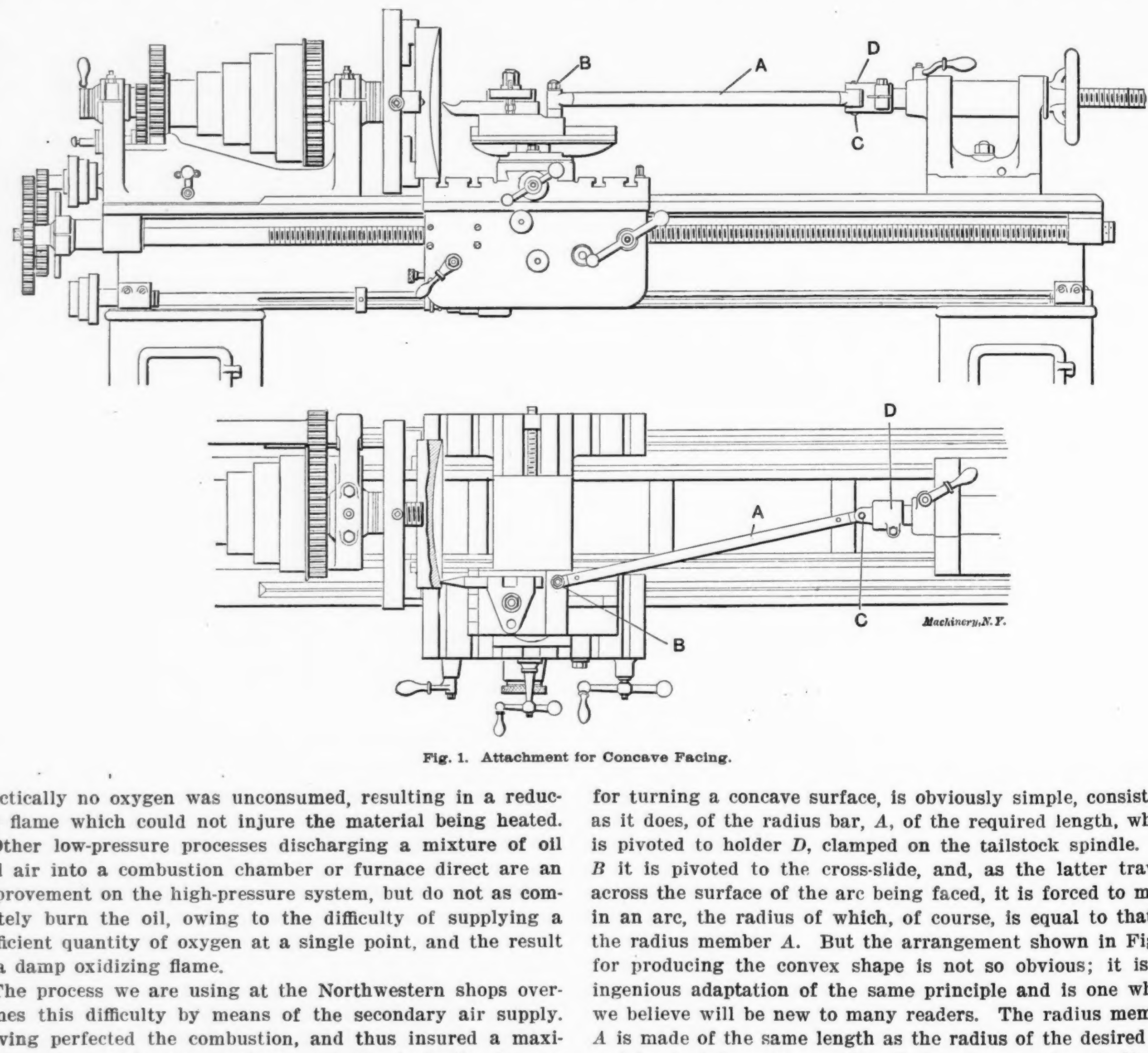
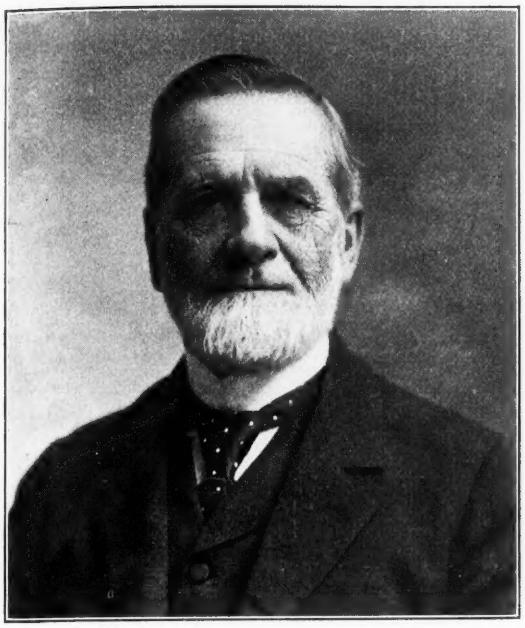


Fig. 1. Attachment for Concave Facing.

for turning a concave surface, is obviously simple, consisting, as it does, of the radius bar, *A*, of the required length, which is pivoted to holder *D*, clamped on the tailstock spindle. At *B* it is pivoted to the cross-slide, and, as the latter travels across the surface of the arc being faced, it is forced to move in an arc, the radius of which, of course, is equal to that of the radius member *A*. But the arrangement shown in Fig. 2 for producing the convex shape is not so obvious; it is an ingenious adaptation of the same principle and is one which we believe will be new to many readers. The radius member *A* is made of the same length as the radius of the desired arc and is pivoted at *I* to a slide, *K*, free to move longitudinally on the lathe bed. The other end of *A* is pivoted to a cross-slide *F*, which moves on a guide, *E*, rigidly secured to the lathe bed. The carriage cross-slide has the roller, *G*, which engages jaws in slide *F*, and hence, as it is fed across the surface of the work, the slide *F* is carried along with the carriage cross-slide. The resultant effect of the movement of *A* is to move the block *K* along the lathe bed, and this movement is transmitted to the carriage by means of the connecting bar, *L*, making the point of the tool describe an arc of which *A* is the radius.

HORACE SILSBY.

The recent death of Horace C. Silsby, of Seneca Falls, N. Y., means the passing away of another man who has been intimately associated with the early industrial development of the country, and in this case with the development of one of



Horace Silsby.

the important branches of machine manufacture. Mr. Silsby was one of the pioneer builders of the steam fire engine, and probably built the first commercially successful engine manufactured in the United States. The earliest steam fire engine which probably had the first paid fire department. Various other builders took up the manufacture of steam fire engines after Mr. Latta, and among these was Mr. Silsby. Appleton's Dictionary of Mechanics, published in 1866, states in regard to the Cincinnati fire department, that steam fire engines had then been in use in Cincinnati for several years and according to the report of the chief engineer of the department had materially reduced the loss from fire. They had not been incorporated into the fire service of other cities, however, though they were used in isolated instances, largely as an experiment.

The father of Horace Silsby was Seth Silsby, a practical blacksmith, who is believed to have been the originator of the chopping ax with a cast-steel blade welded to an iron body. In 1833 the Silsby brothers, of which Horace was one, located at Seneca Falls for the manufacture of cast-steel axes, mill picks, etc. A shop was built for them containing four forges, where the axes were made by hand. Charcoal for welding was brought from a nearby town. The original shop is still standing at Seneca Falls, and is used for a storehouse. It was intended to use water power for grinding and finishing the axes, but the power was found insufficient, and the practice was to grind in the day time and to finish at night. As indicating the primitive commercial methods of those early days, it is interesting to note that the first house owned by Mr. Silby was paid for in axes manufactured by him and his brothers. Later his business interests shifted, and extended into mercantile lines for several years, but in 1847 Horace Silsby, with others, began the manufacture of pumps, stove plate, and regulators for stoves. This line of business increased rapidly and buildings were erected to accommodate it, as the need for them arose, until a large tract of land was covered. In 1853 Birdsell Holly, who was the inventor of an elliptical rotary pump and engine, was admitted into partnership with the firm, the name becoming Silsby, Race &

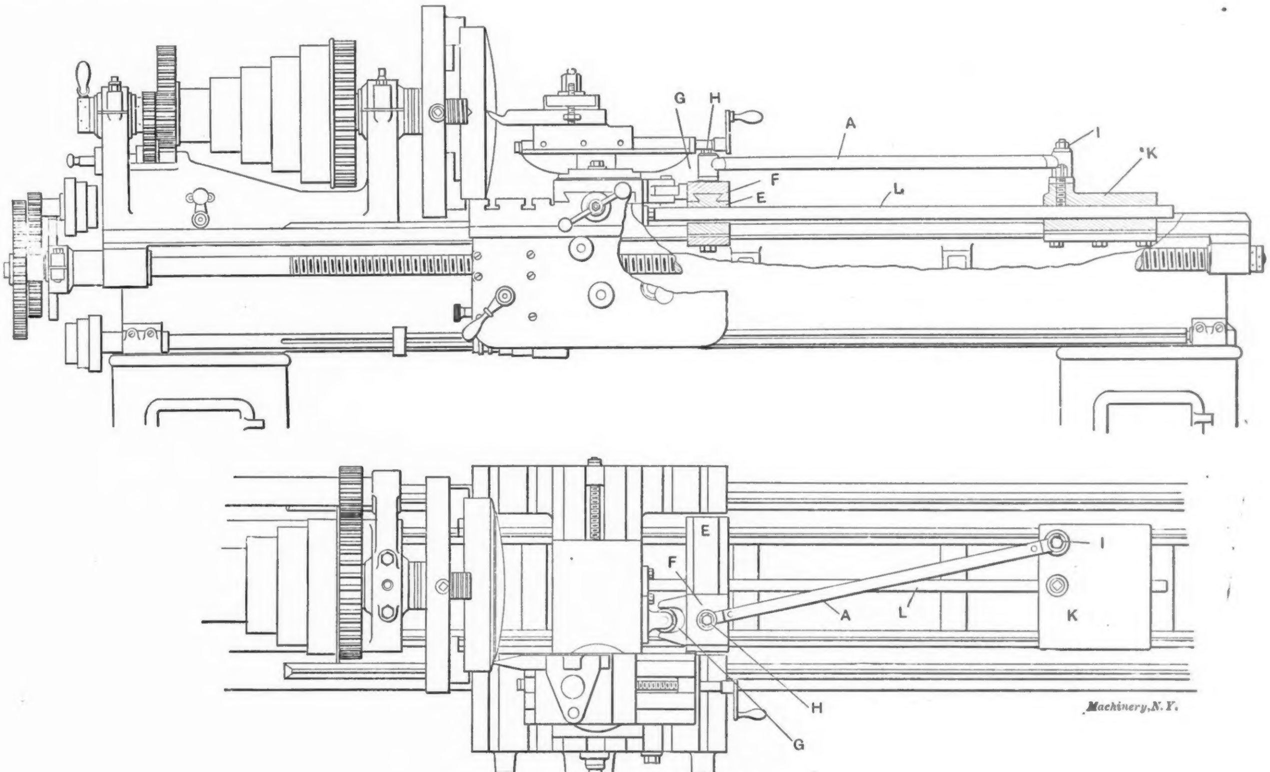


Fig. 2. Attachment for Convex Facing.

is believed to have been constructed in England by John Braithwaite in 1829. Captain Ericcson obtained a medal from the Mechanics' Institute of New York in 1840 for an original design of a steam fire engine which resembled, however, that produced by Braithwaite; but the first steam fire engine to be constructed in America was made by A. & B. Latta, of Cincinnati, and used by the fire department of that city in 1852,

Holly. Holly invented his rotary engine and pump the following year and as the attention of the public was then being attracted by the various experiments in the construction of steam fire engines that were then being carried on at Cincinnati and elsewhere, Mr. Silsby thought the Holly patent could be applied to the steam fire engine; and upon trial it proved successful. In 1856 the first Silsby steam fire engine

was brought out, and was exhibited at the Crystal Palace Exposition in New York; but unfortunately the operation of the machine was not satisfactory and it was taken back to Seneca Falls and consigned to the scrap heap. In 1858 the "Long John" was built, a large and tremendously heavy steamer, which overcame many of the defects of the first machine, and was so far successful that it was in commission in Chicago doing active duty for more than 14 years. The engine was so heavy that its transportation to a fire in rapid time was a matter of considerable difficulty. This machine, however, is justly characterized as a practical and a successful one and rendered more years of service with fewer alterations and repairs than any of the very early fire engines.

In 1855, or at the time the first Silsby fire engine was being constructed, Mr. Silsby bought out the interests of his partners, Mr. Race continuing the stove and regulator business, and Mr. Holly going to Lockport, N. Y., where he developed the well-known Holly water works system. He later became associated with other partners, one of whom, Mr. Mynderse, was an advocate of the piston engine, in preference to the rotary, and one of this type was built. Mr. Silsby advocated the rotary principle, however, and again became sole owner of the business, until his two sons later entered into the enterprise with him. The concern has since become the property of the American Fire Engine Co. and this company is now a part of the International Fire Engine Co., composed of practically all concerns manufacturing fire apparatus.

It is a noteworthy fact that, in spite of the hundreds of patents taken out in this country upon rotary engines, this engine and pump invented by Holly, and placed upon the market by the business enterprise of Silsby, has been the most successful of any of this type. The Silsby engines have given good satisfaction, and the service an engine and pump have to perform in fire fighting is probably better adapted to the rotary type than is almost any other duty. Such machines do not have to be extremely economical, and their work is intermittent; they receive careful attention, and the mechanism of the rotary engine is extremely simple, which is a good feature.

Mr. Silsby was nearly 90 years old at the time of his death, and in the management of the firm with which he was so long associated he always showed appreciation of the worth and genius of his employees, as evidenced by the fact that there were more old men to be found on its pay roll, who had grown gray hairs in its service, than were often to be found, even in old established shops. Mr. Silsby was regarded as the leading citizen of Seneca Falls and through his various enterprises and business connections had done much to promote the interests of his town.

* * *

DEATH OF PROF. REULEAUX.

Prof. Franz Reuleaux, of Berlin, Germany, the distinguished mechanical engineer and professor of engineering, died on August 20, at the age of 76 years. His father was the proprietor of a small machine shop, where Prof. Reuleaux's practical education began. He obtained his technical training at the Polytechnic Institute of Karlsruhe, and later studied at the universities of Berlin and Bonn. After his student days were over, he became manager of a machine shop in Cologne, and then accepted a position as instructor at the Zurich Polytechnic Institute. One of his associates at this Institute was Zeuner, who has become so widely known for his researches in thermodynamics and the mechanism of the steam engine. While at Zurich he began his work in the line of kinematics, for which he is perhaps the best known. As a result of this work he brought out his theoretical kinematics, and only a few years ago issued a second volume upon this subject, called "Applied Kinematics." In developing this subject, he succeeded in establishing relations between the different elements that enter into nearly all combinations of mechanism, so that the subject really became a science susceptible of systematic study. In 1864 he became professor at the Berlin Trade School and four years later was elected to its directorship. In 1879 he accepted an appointment at the Technical College of Berlin, where he remained over 17 years. Of his numerous literary productions, "The Constructor" is his most practical and most widely used work, this having been translated into several languages, and being extensively used in Germany, especially as a text book by students, and as a hand book by engineers.

Besides his work as a professor and investigator, Prof. Reuleaux was active in many ways as a member of different engineering societies, and as public commissioner, appointed to investigate and report upon various engineering enterprises. In 1876 he was sent by the German government to the Centennial Exposition, and in his report upon the machinery exhibition there, he emphasized the inferiority of the work of German machine shops, which he characterized as cheap and poor. This aroused a great deal of discussion, and resulted in a marked improvement in the manufacture of machines in Germany, and was probably the thing that first led Germany to take the important place she now occupies in this field. He again represented Germany at the Chicago World's Fair in 1893, but this time took occasion to compliment his countrymen upon the splendid showing they were able to make in their machine exhibition. Besides his professional duties, he accomplished much in the line of literary work, and was broadly accomplished in many ways other than the particular work to which he devoted most of his time.

* * *

OCTOBER DATA SHEET.

To judge from the great number of tables which have been sent to this office dealing with the problem of squaring mixed numbers, this question must have been occupying the minds of a great many mechanics. Of those we have on hand, the table we publish this month was selected because it contained an extended list of functions useful in many other ways besides in finding the square.

Suppose we wish to find the area of a square the length of whose side is 3 5-64 inches. According to the binomial theorem, "the sum of the squares of two numbers is equal to the square of the first, plus twice the product of the first and second, plus the square of the second." This may be expressed algebraically thus:

$$(a + b)^2 = a^2 + 2ab + b^2$$

In order then to square this quantity, 3 5-64, without reducing it to a decimal, we may add together the square of its first quantity, 3, twice the product of 3 and 5-64, and the square of 5-64. The square of 3 may be obtained mentally, likewise twice the product of the first and second terms; the square of the second term is found in the column of the data sheet headed "square." Written out, as the operation would be performed on paper, it would be expressed thus:

$$\begin{array}{rl} 3^2 & = 9. \\ 2 \times 3 \times 5/64 = 15/32 & = 0.46875 \\ 5/64^2 & = 0.006104 \\ \hline 3\ 5/64^2 & = 9.474854 \end{array}$$

It is not necessary, however, to write all this down. Taking another example, the problem is to square the quantity of 8⅛; all the calculation that needs to be done on paper is given as follows:

$$\begin{array}{r} 8^2 + \tfrac{1}{8}^2 = 64.015626 \\ 2 \times 8 \times \tfrac{1}{8} = \ \ 2. \qquad \\ \hline 8\tfrac{1}{8}^2 = 66.015625 \end{array}$$

In this case the 64.015625 is readily seen to be the sum of the square of the first and second quantities, of which one is always integral and the other always fractional. Their sum may therefore be written at once as a single quantity, the square of the whole number being obtained mentally or from a table of squares, while the square of the fraction is read from the data sheet. To this is added twice the product of the first and second quantities, which amounts to 2 in this case, and may be readily calculated without using a pencil. The result which this method of squaring a mixed number gives is exactly accurate to as many places as the square has been figured to in the table of squares.

Logarithms are given for the various functions to save the usual trouble of reducing the fractions to decimals and then looking up the logarithms in the table.

BORING TOOLS.*

W. J. KAUP.

W. J. Kaup.

In a previous talk on cutting tools I confined myself entirely to one class, namely, planer and lathe tools, and the different conditions under which the best results can be obtained from them. By best results I mean the maximum amount of good work with the minimum amount of energy expended—the ideal for which every good mechanic is striving. I tried to make plain the cardinal points for securing these results, such as proper top and side rake clearance, rigid setting, tool location so that it will not spring into the work, proper relation of cutting wedge to plane of work, etc. All these combine to make the cutting edge the basis of economic production, and economic production means not only least cost in manufacture, but a saving in wear and life of the machine.

I have purposely divided the subject of cutting tools into two separate heads, as there is a recognized distinction between inside and outside turning. The rake and clearance of a tool for inside turning must be different from that used for outside turning, for two reasons: First, because of the contracted and peculiar conditions under which the boring tool works, and second, because of the spring of both tool and

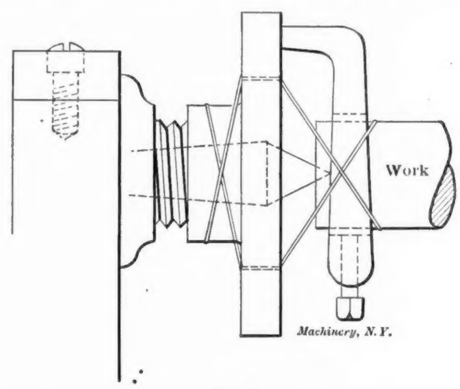


Fig. 1.

work—very serious conditions met with in boring which do not apply in outside turning. The spring of the work is overcome in many cases by using a steady rest to support one end of the work while the other end is held in the chuck, or else is clamped to the faceplate and in addition is sometimes supported by the live center itself. In the latter case it must be laced to the faceplate.

Fig. 1 may serve as a help to some who have found difficulty in keeping the work tight against the center. It shows the faceplate partly unscrewed. The lacing is made fast to a dog or carrier in that position, and the faceplate is then screwed up in place, thereby tightening the thong. Now, unless great care and skill are combined in setting the steady rest in position, the result will be failure because, in boring, the object is to get the bore concentric with the outside and it is a very easy matter to defeat this object by careless setting of the rest. A suggestion as to the way of setting may here be in order. If it is a piece that has already been turned on the outside the centers may be used to good purpose. Keep the live center in the lathe spindle, screw the chuck in position and put the work on centers, as for ordinary turning. Now bring the chuck jaws down to the work and place the rest in position at the dead center end, the work all the while being still on centers. The rest is then opened, the chuck, with the work in it, unscrewed and the center removed. This method will insure accuracy, where it can be used.

If it is a rough piece of work that is to be set, support one end by the dead center, turn a true surface for the jaws of the steady rest and place the same in position while the work is still on the center.

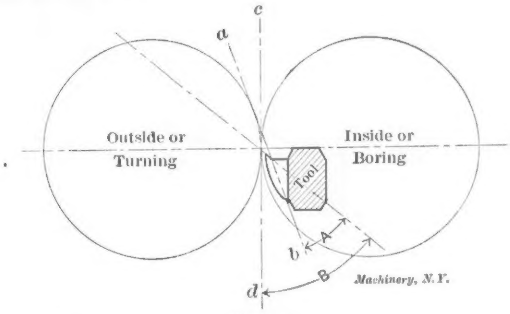


Fig. 2.

Fig. 2 will, I think, prove that the same laws do not hold for both inside and outside turning. The circle on the left represents a cylinder to be turned; that on the right, a hole to be bored, with the tool in position. The lines *a b* and *c d* are drawn tangent with the work at the point where the cutting edge is in contact with the work when turning and boring respectively. On the face of it, it would seem that one vertical line should answer for both conditions, but not so, for in turning we are enabled to set the cutting edge of the tool above the center of the work, hence changing the position of the lines and getting a finer cutting wedge. The angle *A* is

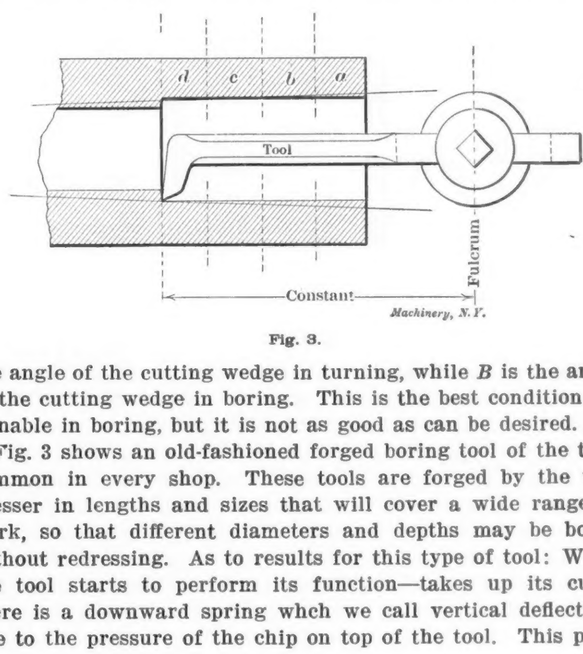


Fig. 3.

the angle of the cutting wedge in turning, while *B* is the angle of the cutting wedge in boring. This is the best condition obtainable in boring, but it is not as good as can be desired.

Fig. 3 shows an old-fashioned forged boring tool of the type common in every shop. These tools are forged by the tool dresser in lengths and sizes that will cover a wide range of work, so that different diameters and depths may be bored without redressing. As to results for this type of tool: When the tool starts to perform its function—takes up its cut—there is a downward spring whch we call vertical deflection, due to the pressure of the chip on top of the tool. This pres-

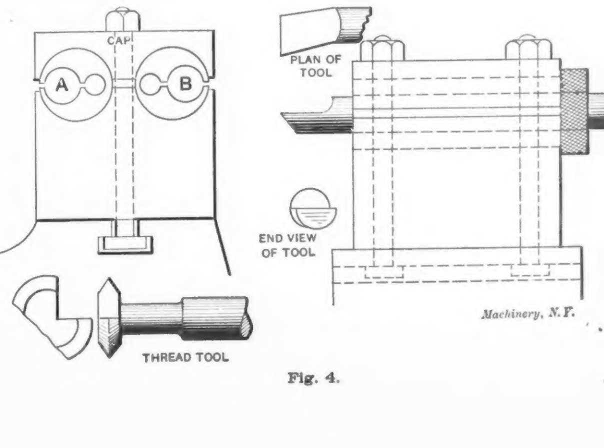


Fig. 4.

* Abstract of a lecture by W. J. Kaup, instructor in charge of the machine department, Pratt Institute, Brooklyn, N. Y. The machine shops of Pratt Institute are equipped with a great variety of modern machine tools and under Mr. Kaup's direction the course in shop practice has been developed to a high state of efficiency, and is conducted as far as possible to conform with the methods used in the best shops doing commercial work. Each year Mr. Kaup gives a series of lectures to young men, many of whom are machinists, and he has often found that instruction in some of the simpler and more elementary points connected with the subject was more urgently needed than was information upon the more complicated details. The abstract published herewith is from one of a series of lectures upon cutting tools.

sure is nearly constant throughout the entire length of the cut and does not vitally affect the accuracy of the work, particularly since there can be a slight vertical movement of the tool without appreciably changing the diameter of the surface being bored. This is not the case, however, with the lateral pressure on the boring tool, which pressure, being at right angles to the cutting edge, deflects the tool away from the work more and more as the cutting edge dulls, thereby changing the angle of motion of the tool constantly. The result is

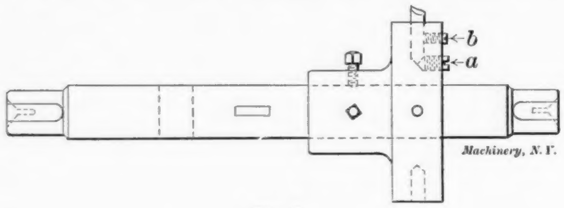


Fig. 5.

a conical hole, and much time is lost in taking repeated cuts to get the bore parallel. This type of tool, therefore, does not prove economical, although the outward or lateral pressure will vary somewhat with the shape of the tool and the way in which it is dressed.

If the front or cutting edge makes an acute angle with the work, the lateral pressure is considerable; but if the cutting edge is at right angles to the work there is less tendency to deflect the tool in a sidewise direction. In the latter case, however, as the cutting edge wears away and the tool becomes

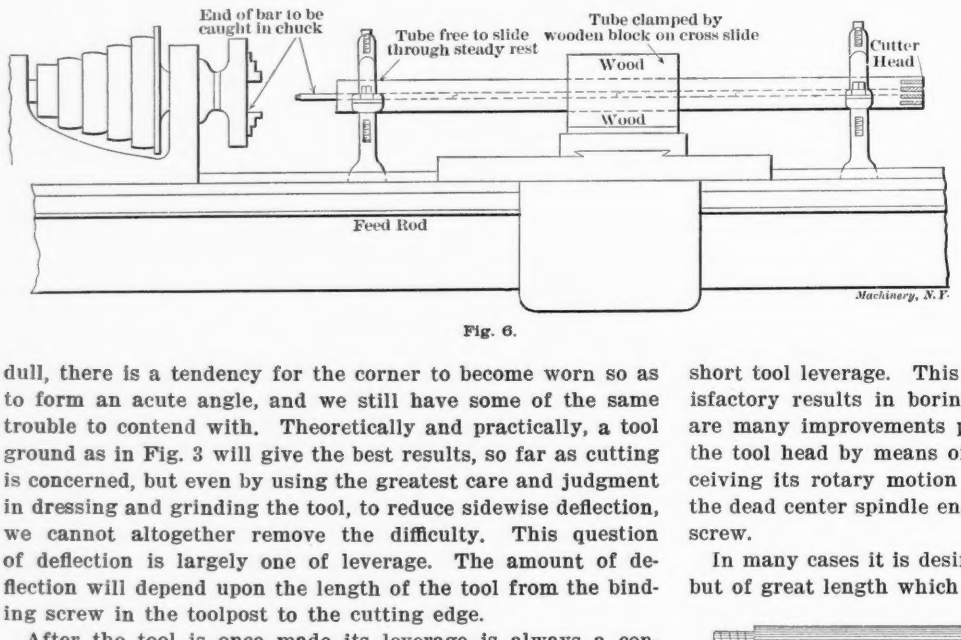


Fig. 6.

dull, there is a tendency for the corner to become worn so as to form an acute angle, and we still have some of the same trouble to contend with. Theoretically and practically, a tool ground as in Fig. 3 will give the best results, so far as cutting is concerned, but even by using the greatest care and judgment in dressing and grinding the tool, to reduce sidewise deflection, we cannot altogether remove the difficulty. This question of deflection is largely one of leverage. The amount of deflection will depend upon the length of the tool from the binding screw in the toolpost to the cutting edge.

After the tool is once made its leverage is always a constant quantity, as indicated in Fig. 3, since the tool must always be placed in approximately the same position in the toolpost. It will, therefore, deflect as much in boring a short hole as in boring a long one, assuming the cutting edge to be in the same condition in each case. The longer the tool the greater the deflection, for the tool is a cantilever the deflection of which is increased eight times when its length is doubled. From this we can readily see how important is this consideration of leverage, and how desirable it is to have the boring tool adjustable so that it need project from the point of support only so far as is necessary to bore the full depth of hole required. The mechanic should try to overcome the difficulty due to leverage by devising ways and means for making the tool adjustable. Many schemes are open to the thinking mechanic.

Fig. 4 will give an idea for a tool holder and for different tools which are inexpensive and at the same time meet the above requirement. The holder gives at all times the greatest rigidity and allows the use of the largest size of tool possible for any particular work. It also enables the operator to vary the leverage to suit each particular hole.

The holder consists of a rectangular block of cast iron in which two holes are bored, one on each side of the center, and on a plane with the lathe center, and extremely close to the edges. After this it is sawed in two through the center of the holes, the top forming the cap. The hole may be made any standard size: 1¼, 1 3-16, 1½. Into these holes are fitted sleeves or the drill rod itself, although the sleeves give wider range of size of boring tool for each holder, by having a number of sleeves with different standard size holes. The tool fits in either *A* or *B* of the sleeves. If the tool is to be used in *A*, a solid piece is inserted in *B*, so as to give a support for the cap to be clamped against. One end of the sleeve is knurled to allow for thumb and finger adjustment in raising and lowering the tool. Ordinary drill rod is used, filed down to a flat surface at the end, as shown. When heating for the tempering process, set over the filed end by a blow of a hammer for clearance. A tool nearly the size of the hole to be bored may be used. For instance, an 11-16-inch tool could easily be used to bore a ¾-inch hole. The thread tool in this type is of the greatest advantage in that the tool is always level—the requisite for a true-angle thread.

The good features of this type of tool are: first, it saves in expense in forging; second, it saves time in grinding and setting, and in boring a true hole; third, it requires less skill and judgment in getting results. As the work increases in weight and size, and it is not practicable to clamp either on faceplate or chuck, the boring bar is substituted, in which case former conditions are not encountered.

Many styles of boring bars are used, the one shown in Fig. 5 being possibly one of the simplest type. In the boring bar head you have to consider only the proper cutting edge of the tool; and attention is to be called chiefly to the method of setting out the tool for increasing the depth of cut. The tool itself has a wedge end and the set screw *a* a conical end bearing against it, thus forcing the cutter out as the screw moves in. The binding screw *b* comes in contact with a flat side filed or ground on the cutter. Heads of different sizes are made to fit the bar, to suit holes of different diameters, insuring a short tool leverage. This type of bar was used with very satisfactory results in boring fields for 5 H. P. motors. There are many improvements possible in this bar, such as feeding the tool head by means of a screw carried in the bar and receiving its rotary motion by the use of a stationary gear on the dead center spindle engaging with a gear on the end of the screw.

In many cases it is desirable to bore holes of small diameter but of great length which extend through the entire length of

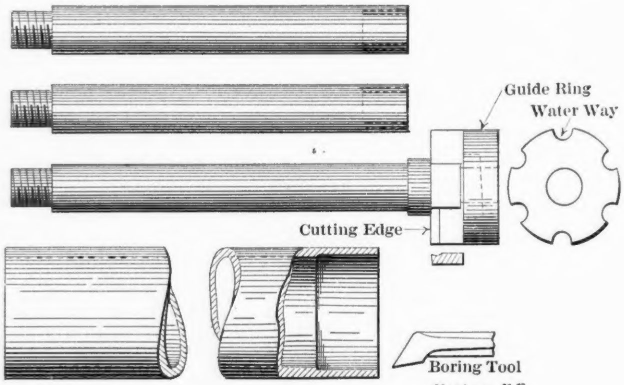


Fig. 7.

a tube, such for example, as core barrels for rock drilling, where the tube is from 10 to 14 feet long, and as small in some cases as 2 inches in diameter.

Fig. 6 will give an idea of the method by which such holes may be bored with very satisfactory results, and in Fig. 7 the boring bar is shown in detail. The bar is made up of sections, say 3 feet long, each so constructed that they can be joined together into one long bar. The work is supported in the

lathe by two steady rests and is clamped to the carriage of the lathe by means of wooden clamps, specially constructed to suit each individual case. The steady rests are only used to guide and support the tube as the carriage advances, carrying the tube with it. The end of the tube is first bored to a depth of about 2 inches with the ordinary boring tool, and made the required size. The bar is then inserted until the cutter head reaches the bored end of the tube which the guide ring on the outer end of the head should fit nicely. The tube is then clamped to the carriage, supported by the steady rests, and the

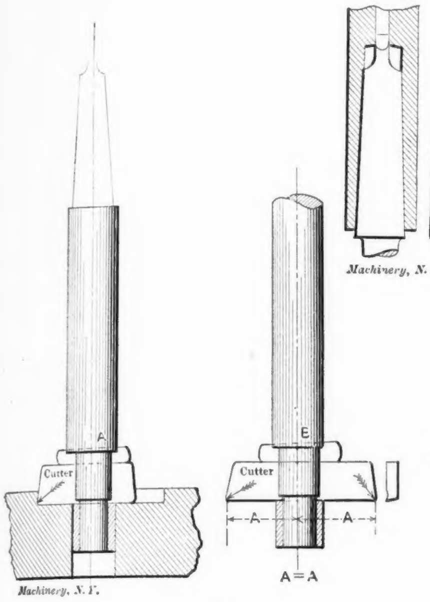


Fig. 8.

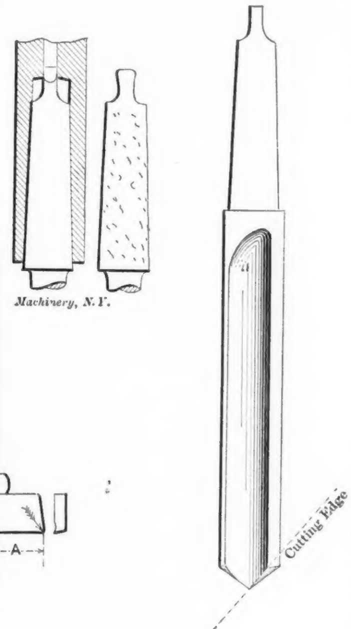


Fig. 9.

other end of the bar is held by the lathe chuck. Allowance should be made for the tube to travel a distance equal to the entire length of one of the boring bar sections. When the tube is advanced this far one of the sections of the bar is unscrewed and laid aside and the chuck engages the end of the next section, and so the work proceeds until completed. It should be observed that the face only of the tool should be used as a cutting edge, while the outside acts simply as a guide.

In Fig. 8 is another common type of boring tool more generally known as a counterbore, but it is used to a great extent as a boring tool proper. It is important that there be two cutting edges, to balance or equalize the resistance and obtain round holes. Too often we find it used as a fly cutter, without the advantage of speed which is essential to the success of a fly cutter, as shown at *A*, Fig. 8. When the cutter takes up its work, the resistance due to the chip tends to push it away, and this strain is carried entirely by the bushing or guide pin of the bar, which results in the lapping out of uneven wearing in the hole. This means only one thing — the counterbored part will neither be round nor of the size intended.

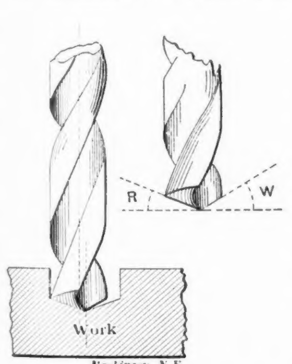


Fig. 10.

When a piece is irregular in shape, and in cases where the boring mill is not available, the drill press must be converted into a vertical mill. The work is clamped to the table of the drill press and the boring bar is carried by the drill spindle at its upper end and is guided at its lower end by a bushing in a hole at the center of the table.

By far the most important boring tool in any shop is the drill, and experience has proven that with no other type of boring tool have there been so many disappointing results as with drills. It must be patent to every mechanic that the cutting edges of the drill should have a uniform angle with the longitudinal axis of the drill and they should be properly cleared or backed off and should be of equal length. The angle for iron and steel is from 59 to 60 degrees, and for brass from 75 to 76 degrees—a greater angle in the latter case because of the tendency of the drill to hook into the softer metal, especially as the drill is about to break through the work. The angle of the point is a sure indication of the lip clearance, as illustrated at *a*, *b*, *c*, in Fig. 11. Too much clearance, as shown by *b*, will produce irregular shaped holes.

Fig. 10 indicates the results of improper grinding. One lip is much longer than the other, hence the two edges are at different cutting angles and only one lip cuts, making it impossible to drill holes anywhere near the size. Another thing that gives considerable trouble is carelessness about the point of the drill. It must be sharp or else no matter how keen the outer parts of the cutting lip may be the drill will not work without considerable pressure which in extreme cases tends to crush the drill. Another small thing but very important to observe is the starting of the drill. If it runs out of center before it begins to cut the full diameter of the hole

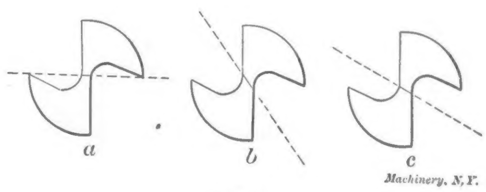


Fig. 11.

the drill should be drawn over by gouge-chiseling the countersunk portion. This is more important in holes that are to be rebored, for it is desirable that equal metal should be left on all sides for the next succeeding bar or cutter. In every instance where accurate drilling is desired it is advisable to drill a small hole first as the large drill will follow more accurately. It also relieves the point of the large drill from acting as a pivot, and a hole of more exact diameter is obtained.

Fig. 9 shows a farmer or straightening drill used successfully on cored holes. This is also available as a cannon bit for lathe work, by boring the work to a size just deep enough for the drill to enter, and by using the tailstock feed.

Another small thing that defeats good work, and the chance for promotion of many mechanics, is shown in the upper left-hand corner of the same illustration, where a little speck of dirt or a small chip came between the drill shank and the socket and made a depression in the drill shank every time the latter was driven up into the socket until the shank had a pock-marked appearance. It is just as important that the shank of the drill should fit the socket of the drill press as it is to have the center of lathes fit in their respective places.

TABLE OF FEEDS AND SPEEDS FOR DRILLS.

Size of Drill.	Rev. per Min.	Feed per Rev.	Feed Rev. per M.
1/4"	380	.005	200
1/2"	180	.007	143
3/4"	110	.010	100
1"	75	.014	75

The accompanying table shows the speeds at which the drills may be run. As to the feeds remember that this depends entirely on the speed at which the drill rotates, and that to get the best results with a drill the highest speed possible for each particular size of drill is essential.

* * *

The building of the great bridge over the Zambesi River at the Victoria Falls on the Cape-to-Cairo Railway has attracted much attention to these wonderful falls and to their capacity for future development. The Zambesi River is a mile wide above the falls and the cataract is 420 feet high. In short, the Victoria Falls have fully twice the volume and more than twice the height of Niagara Falls and it is estimated that the enormous power of these falls approximate 35,000,000 horse power, as against 7,500,000 horse power for Niagara. It is not improbable that this great water power will be employed for the electrification of all the immediate portions of the Cape-to-Cairo Railway, and it is not impossible, with the improvements of the electrical transmission of energy, that a generation hence may see the completion of this great railway project and its complete electrification.

SPIRAL GEAR CUTTING IN THE NATIONAL ELECTRIC COMPANY'S SHOPS.

The building of the motor-driven compressors for use with the well-known Christensen air-brake equipment for street railway cars forms a considerable part of the business of the National Electric Co., of Milwaukee, Wis., and was, in fact, the sole business of the concern in the beginning. It is, of course, very important that these air compressors, which are located beneath the floor of street cars, should run with a minimum of noise and vibration. They are electrically driven by small motors running at high speeds, hence freedom from vibration is a condition difficult to obtain with ordinary spur gears; but spiral gears seem to have solved this difficulty of construction in the Christensen compressors. These gears and pinions are made with right- and left-hand spirals, and are fastened together in pairs so as to form the balanced construction which is characteristic of herring-bone teeth.

In cutting the spiral gears and pinions they are mounted in strings on arbors, eight or ten being mounted on an arbor. The pinions are cut on an ordinary Kempsmith universal mill-

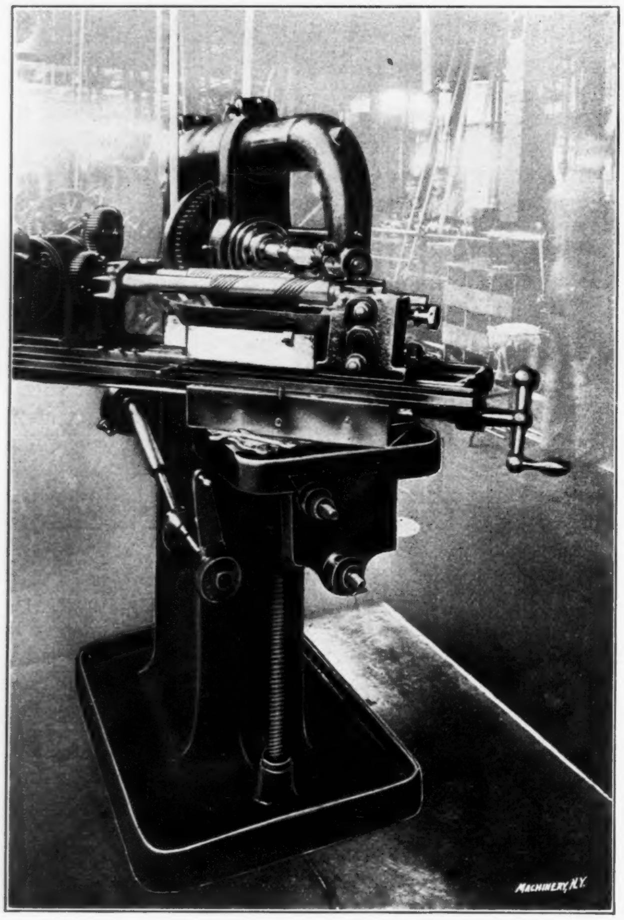


Fig. 1. Cutting two Spirals at once.

ing machine which, however, has a special attachment to the dividing head so that two arbors full of pinions may be cut at once. Attached to the side of the dividing head, as indicated in Fig. 1, is an auxiliary spindle geared to the dividing head spindle by an intermediate gear so that both will turn in the same direction, of course. The milling cutters are mounted on the machine spindle at such a distance apart as permits them to work on both arbors at the same time when the table is swung around to the required angle. It is evident that the headstock and footstock for one arbor is offset so as to stand back of similar planes of the other, thus bringing the pinion blanks in the same relative positions to the cutters.

Fig. 2 shows another Kempsmith knee-type milling machine rigged for cutting the teeth of the spiral gears. In this case the gears are attacked from the side, an attachment being provided which is bolted onto the vertical face of the frame and which carries the arbor for the cutters. A spur gear is mounted on the end of the machine spindle, and meshing with it is a spur pinion mounted on a short shaft together with a bevel gear, which latter transmits motion to the auxiliary spindle carrying the cutter. The cutter is located in line with the axis of the main spindle, so that as the attachment is set at various angles to cut the required spirals its central position is not changed. When two cutters are used for roughing, they are located, of course, at equal distances on each

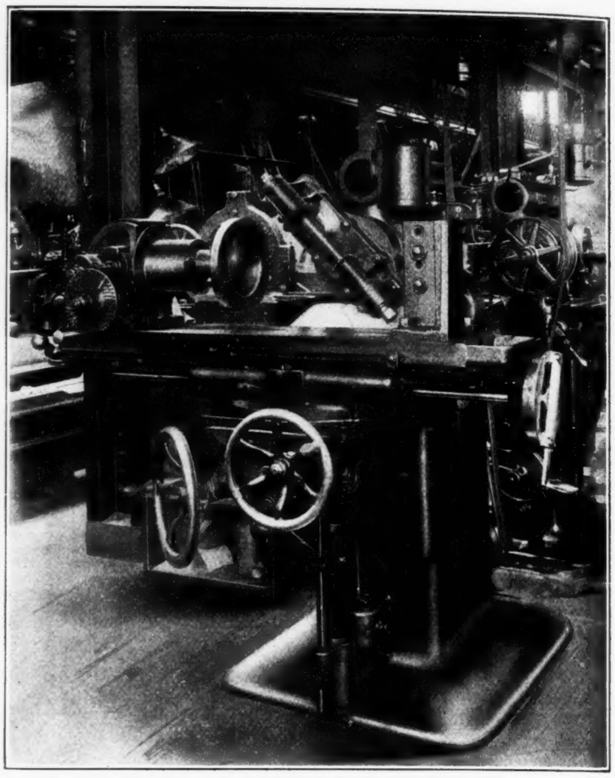


Fig. 2. Spiral Cutting Attachment for Milling Machine.

side of the central line. Fig. 2 shows a front view of the attachment and Fig. 3 a side and rear view of the machine at work cutting the teeth on a string of ten gears mounted on an arbor. The National Electric Co. has found these attachments to be very effective in reducing the cost of spiral gear cutting.

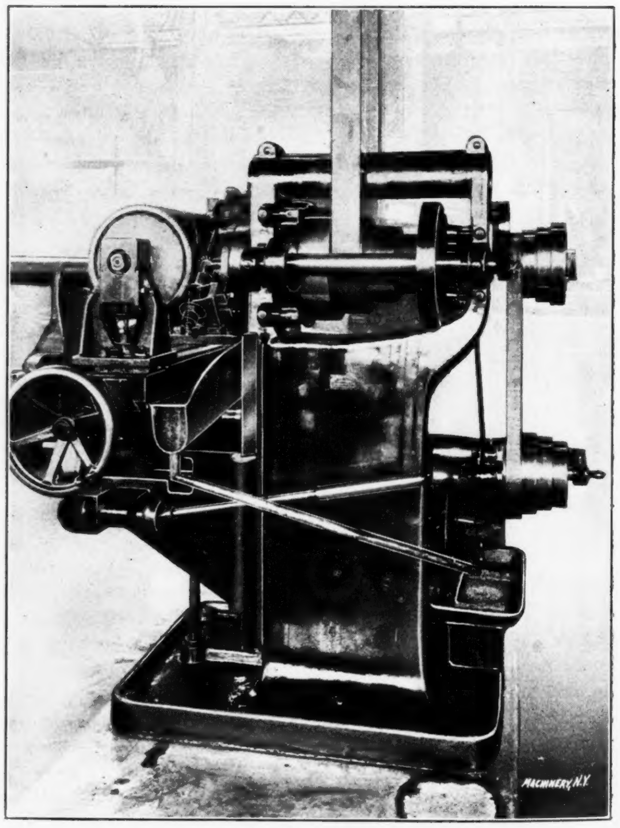


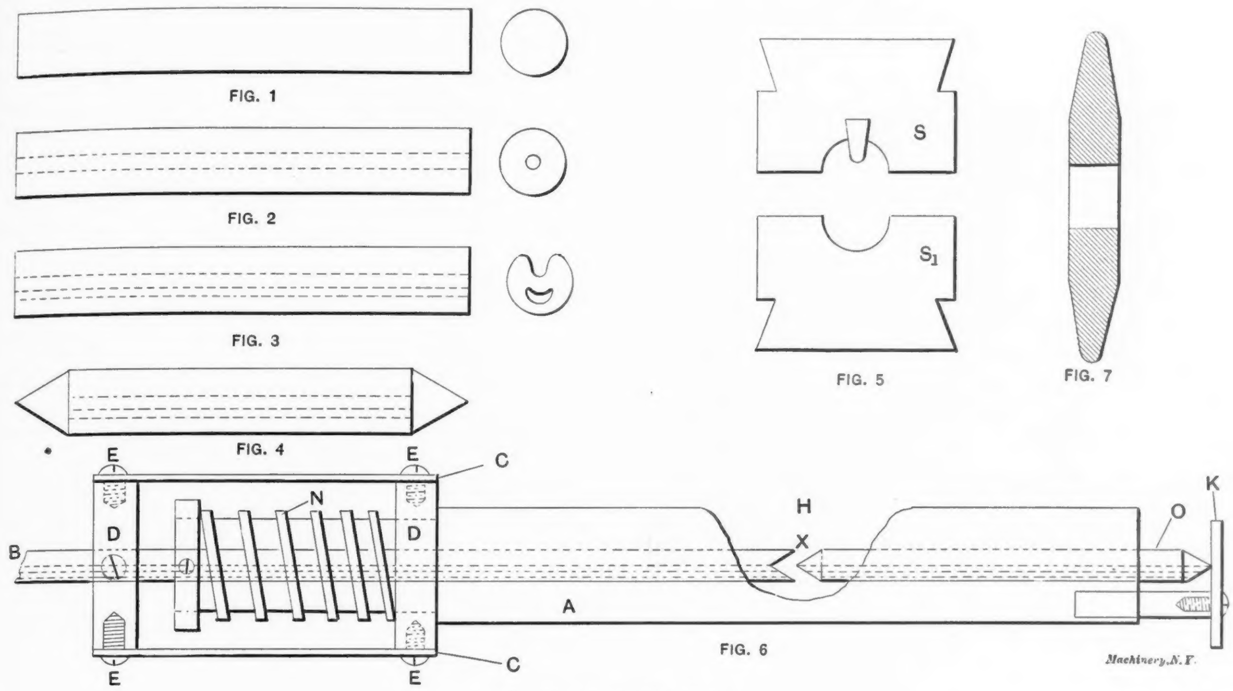
Fig. 3. Spiral Cutting Attachment in Operation.

CONSTRUCTION OF A DEEP-HOLE DRILL.

E. W. NORTON.

In the making of the type of drill shown in the illustrations we differ somewhat from that described by Mr. Eckelt in the February issue; we omit the special gage and use a plain, everyday micrometer, with the end of the spindle pointed a little less than the included angle of groove in drill, which is about 38 degrees.

In making these drills half-inch Novo stock is cut into lengths of about 4 inches (for small calibers) Fig. 1; then a hole is drilled through the center, Fig. 2; next it is heated and struck up in dies S S_1, Fig. 5, which operation forms a groove and forces the center hole, as in Fig. 2, below the center into a crescent shape, as shown in Fig. 3. It is then "pack annealed"; then both ends are pointed to 60 degrees, Fig. 4, and turned in a lathe on female centers to within 0.010 inch of finished size, this being allowed for grinding. After being pointed it is held in a milling machine vise and the groove is finished with the cutter, Fig. 7, a depth equal to one-half the finish diameter, plus 0.005 inch, minus the amount to be ground out of the groove, which varies according to the caliber of the drill and the grade of steel used. In short, the bottom of the groove must be dead central when drill is finished.



The Way the Drill is Made.

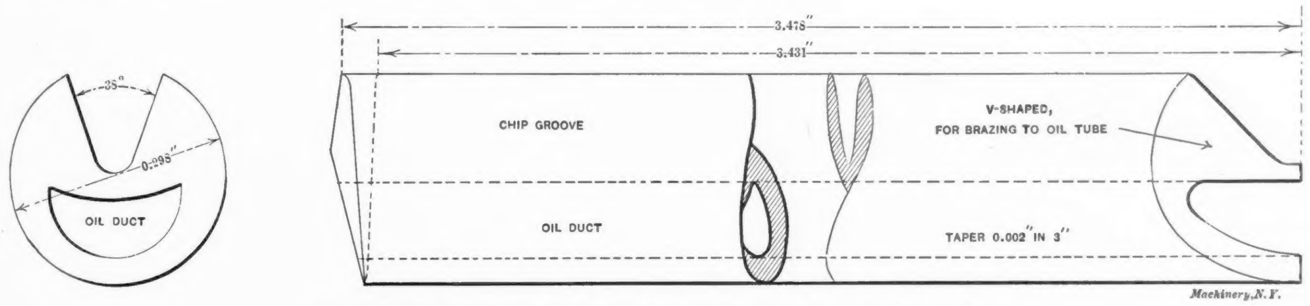


Fig. 8. Detail of Deep-hole Drill, 0.303 Caliber, Enlarged four times Natural Size.

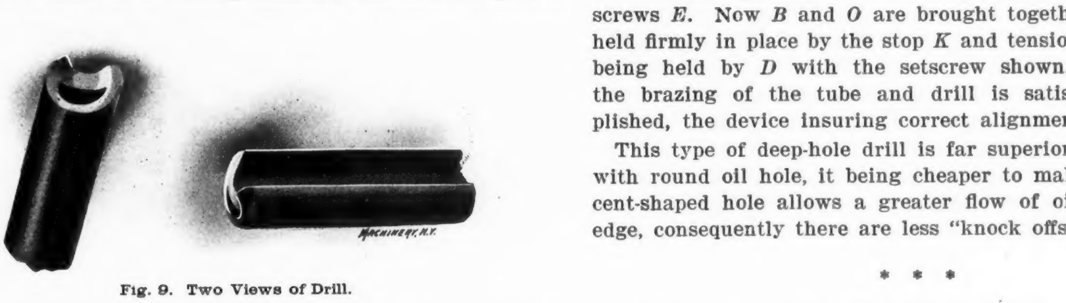


Fig. 9. Two Views of Drill.

The drill is hardened two-thirds of its length, the outside and groove are ground. It is now placed in brazing fixture, Fig. 6. A is the body of suitable length with a hole drilled through it to accommodate all sizes of drills under ½-inch. B is a steel tube a little under the size of the drill; a groove is rolled the whole length of B to conform with the shape of the groove in the drill. A V-groove is milled in the end of the tube to fit the back end of the drill, which is milled off wedge shape to fit. Collars D D are bored out, one a loose fit on A, the other on B.

The drill O is placed in the end of A, then B is inserted into the other end of A. The two are brought together at the opening H, which is milled out to allow point X to come in contact with the flame. A small wooden plug is fitted into O and B to prevent the brazing stopping up the oil-hole. A small piece of silver solder is placed between the two ends; then the swinging stop K is moved into place against the end of drill O; collars P P are held together by strips C and screws E. Now B and O are brought together at X and are held firmly in place by the stop K and tension of spring N, B being held by D with the setscrew shown. With this rig the brazing of the tube and drill is satisfactorily accomplished, the device insuring correct alignment.

This type of deep-hole drill is far superior to the old drill with round oil hole, it being cheaper to make, and the crescent-shaped hole allows a greater flow of oil at the cutting edge, consequently there are less "knock offs" for the driller.

* * *

Baldwin Locomotive Works built 220 locomotives during the month of August, and up to September 1 there had been 1,400 built during the year of 1905. This enormous production is at the rate of nearly 7 per day for each working day of the year.

ITEMS OF MECHANICAL INTEREST.

NOVEL BAND SAW SHARPENING DEVICE.

Automatic machines for filing band saws in the same manner as would be done by hand are well known, but the method employed by Mr. J. J. Rexroth, 54 Adams Street, Chicago, Ill., is one that will undoubtedly be new to most of our readers. The device consists of a wooden frame, shown on the saw table in Fig. 1, in which is mounted a tapered steel cutter resembling a tapered reamer. The teeth, however, are end cut

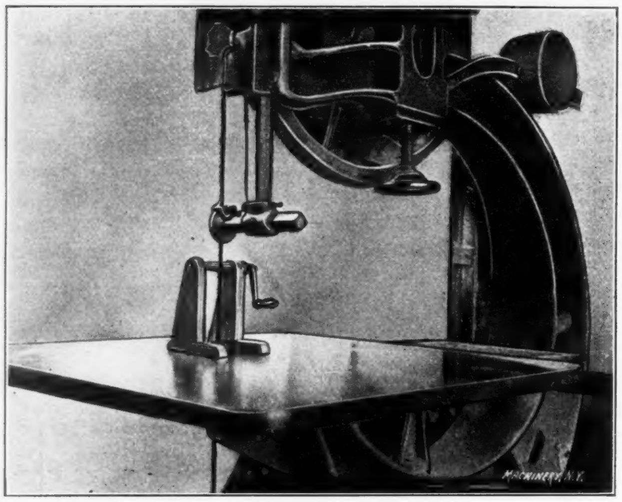


Fig. 1. Device for Sharpening Band Saw.

considerably, as will be seen in Fig. 2. The device is used by setting it on the saw table, as shown, and bringing the teeth of the saw blade to engage with the teeth of the cutter, sliding the latter sideways until the proper pitch to mesh the saw is obtained. Then by turning the crank handle so as to drive the saw upward in a direction opposite to that in which it runs when sawing, the teeth of the cutter remove thin shavings from the under face of the saw blade teeth and sharpen it with great rapidity. Running the saw around two or three

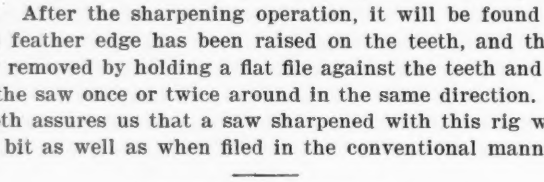


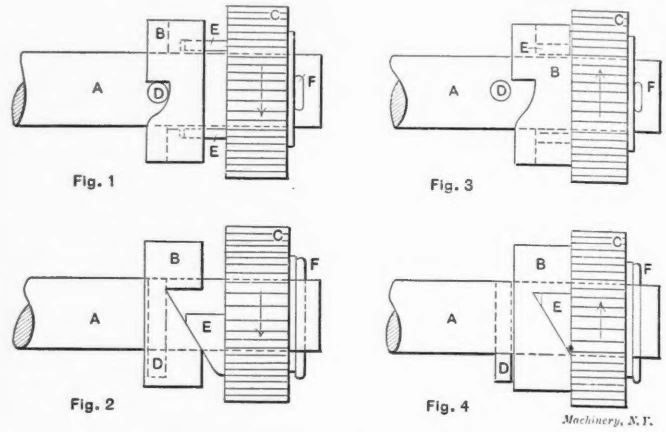
Fig. 2 Cutter for Device shown in Fig. 1.

times is enough, ordinarily, to sharpen it. The cutter tooth acts as a pinion on the saw tooth as they approach, gums the saw as it passes the center, and sharpens it as they draw apart. After the sharpening operation, it will be found that a fine feather edge has been raised on the teeth, and this is easily removed by holding a flat file against the teeth and running the saw once or twice around in the same direction. Mr. Rexroth assures us that a saw sharpened with this rig works every bit as well as when filed in the conventional manner.

NOISELESS RATCHET.

Considerable ingenuity has been displayed in the design of the ratchet mechanism of lawn mowers to make them noiseless, it having been the aim of several inventors to produce a lawn mower in which there will be an absence of the characteristic clicking noise when the machine is reversed. In one type of machine this part of the mechan'sm consists of a pinion containing a three-toothed internal ratchet. The pinion is loosely mounted on the cutter shaft and a hole is provided through the shaft in the plane of the three-toothed ratchet. Through this hole works a cylindrical pin made of such length that it can work back and forth through the shaft when the pinion is reversed. The moment the pinion starts in the forward or cutting direction the radial side of one of the ratchet teeth engages with the pin and drives the cutter shaft.

Another scheme is that shown in the accompanying cut in which *A* is the cutter shaft, *B* is a loose collar on the shaft and *C* is a toothed pinion. Collar *B* has two notches on the left side for engagement with the pin *D*. The left side of pinion *C* is provided with two teeth, *E*, which engage in recesses cast on the right side of collar *B*. The driving direction of pinion *C* is in the direction of the arrow, Figs. 1 and 2. From this it is evident that motion is transmitted from the pinion through teeth *E* to collar *B*, the teeth *E* driving with their inclined sides. This action, of course, forces collar *B* over to



A Noiseless Ratchet as applied to a Lawn Mower.

the left so that one of the notches on the left side are brought into engagement with the pin *D*. Reversal of movement causes the collar to slide off pin *D* and approach the pinion, thus throwing the driving mechanism out of engagement with the shaft *A*, as indicated in Figs. 3 and 4. During the reverse movement there is no reciprocating action of the clutches, hence there is no clicking sound. The moment, however, that pinion *E* is again reversed to drive, collar *B* lags behind, the result of which is that it is forced over to the left and brought again into engagement with pin *D*, thus transmitting motion to the cutter shaft. The cotter *F* takes the end thrust of *C* when driving in the cutting direction.

TABLE FOR SETTING PROPORTIONAL DIVIDERS.

The table given below was contributed by Mr. Herman Jonson, New York, who writes:

I send you herewith a copy of a page from my notebook—a table of setting for proportional dividers, which I have found useful. While I do not believe in the use of proportional dividers on exact work, that is, a drawing which must be scaled, they are all right to lay in the outlines of machinery in building plans and in catalogue work. In addition to the setting which the table gives, it also enables one to see at a glance what proportion a scale drawing is. Referring to the table we see that ⅜-inch scale is 1-32 of 12 inches (full size).

						B	*A*								
		3/32″	1/8″	3/16″	1/4″	3/8″	1/2″	3/4″	1″	1 1/2″	2″	3″	4″	6″	12″
	3/32″	1	3/4	1/2	3/8	1/4	3/16	1/8	3/32	1/16	3/64	1/32	3/128	1/64	1/128
	1/8″	1 1/3	1	2/3	1/2	1/3	1/4	1/6	1/8	1/12	1/16	1/24	1/32	1/48	1/96
	3/16″	2	1 1/2	1	3/4	1/2	3/8	1/4	3/16	1/8	3/32	1/16	3/64	1/32	1/64
	1/4″	2 2/3	2	1 1/3	1	2/3	1/2	1/3	1/4	1/6	1/8	1/12	1/16	1/24	1/48
	3/8″	4	3	2	1 1/2	1	3/4	1/2	3/8	1/4	3/16	1/8	3/32	1/16	1/32
a	1/2″	5 1/3	4	2 2/3	2	1 1/3	1	2/3	1/2	1/3	1/4	1/6	1/8	1/12	1/24
b	3/4″	8	6	4	3	2	1 1/2	1	3/4	1/2	3/8	1/4	3/16	1/8	1/16
	1″	10 2/3	8	5 1/3	4	2 2/3	2	1 1/3	1	2/3	1/2	1/3	1/4	1/6	1/12
	1 1/2″	16	12	8	6	4	3	2	1 1/2	1	3/4	1/2	3/8	1/4	1/8
	2″	21 1/3	16	10 2/3	8	5 1/3	4	2 2/3	2	1 1/3	1	2/3	1/2	1/3	1/6
	3″	32	24	16	12	8	6	4	3	2	1 1/2	1	3/4	1/2	1/4
	4″	42 2/3	32	21 1/3	16	10 2/3	8	5 1/3	4	2 2/3	2	1 1/3	1	2/3	1/3
	6″	64	48	32	24	16	12	8	6	4	3	2	1 1/2	1	1/2
	12″	128	96	64	48	32	24	16	12	8	6	4	3	2	1

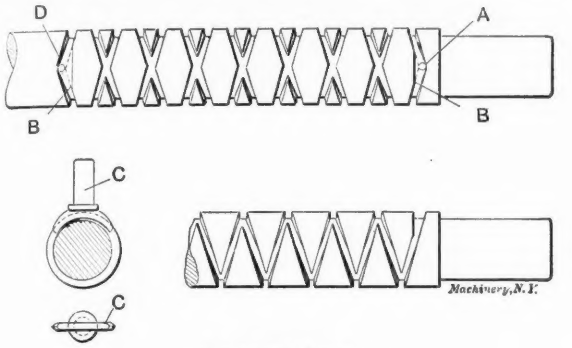
Horizontal line of figures at top represents inches per foot. Vertical line of figures at left represents inches per foot.

Example: A drawing is made ¾ inch to one foot, and we wish to make it ½ inch to one foot. Following line (*a*) ½ inch to the right till we come to (*A*) ¾ inch, we obtain a setting of 2-3 on the proportional dividers. It will be noticed that this may also be read on line (*b*) ¾ inch to the right until we come

to (*B*) ½ inch, and we obtain setting 1½. But as we have no setting 1½ on the dividers, we reduce 1½ to the improper fraction of 3-2, and inverting same we have 2-3 as the required setting.

CUTTING A TRAVERSE SCREW.

Mr. James F. Coyne, of East Providence, Rhode Island, sends us a description of a "traverse screw" which he has recently cut. It consists, as may be seen from the line engraving below, of a shaft which is cut with a right-hand and left-hand thread, both starting from the same drill hole *A* and ending in the same drill hole *D*. *C* is a guide or traveler which is adapted to move easily in the thread. If the shaft revolves continuously the traveler will be carried toward one end of the shaft. It will be noticed that at point *B* at each end of the screw, the thread shown by the dotted lines has been cut away



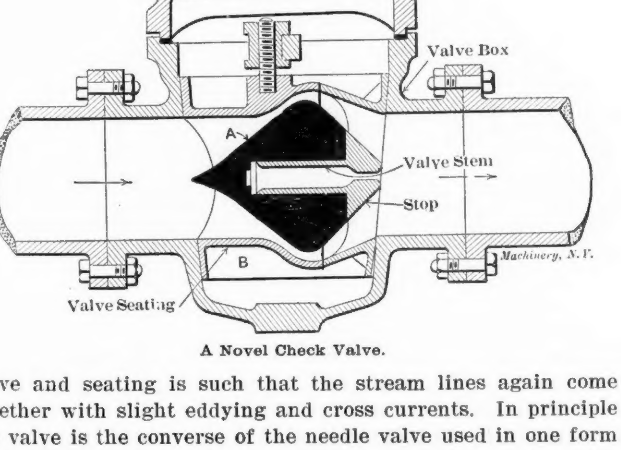
A Traverse Screw.

to form a widened recess. When the traveler reaches the end of the right-hand thread this recess gives it room to shift its position slightly and engage the beginning of the left-hand thread; which, in turn, carries it to the other end of the screw. The continuous motion of the shaft will then give a continuous reciprocating motion to the traveler *C*. This may be used for operating any light mechanism desired.

This device of course is not new, as the principle has been used in textile work for many years on spooling, and other machinery. There are probably few machinists, however, who have ever had occasion to cut a screw of this type, so the job is unusual enough to be interesting.

CHECK VALVE FOR UNOBSTRUCTED FLOW OF LIQUID.

The accompanying cut shows a peculiar valve box which illustrates a principle of construction employed in check valves by an English concern. The valve and seating are so designed as to give an approximately straight flow to the liquid and to make the area of the opening equal to that of the pipe in which the valve is inserted. It will be noticed that the valve *A* is of peculiar conoidal form, which splits the stream lines and causes the liquid to flow around it with a gently accelerated motion outward. The shape of the anterior part of the



A Novel Check Valve.

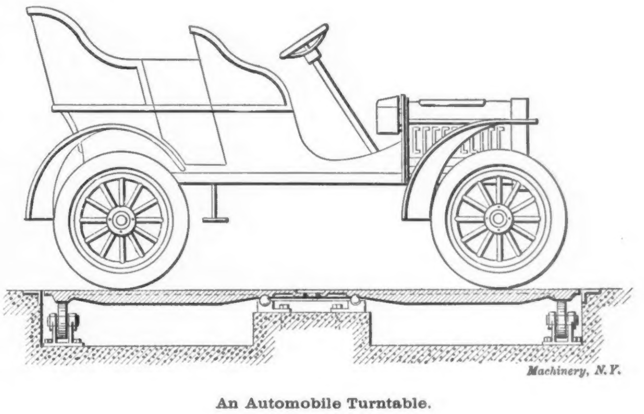
valve and seating is such that the stream lines again come together with slight eddying and cross currents. In principle the valve is the converse of the needle valve used in one form of the Pelton wheel governor. The construction of the valve box in this case is such that the valve and seating may be removed from the pipe line without disturbing the connections; this being done by removing the top, *C*, and pulling the conical plug containing the valve out of the case.

DEVICE FOR TESTING AIR BRAKES.

An interesting train-pipe device for testing air brakes has been patented, by means of which an engineer can test the brakes of a train and ascertain whether all connections have been properly made, and the cocks opened. The test device is attached to the last car of the train and is connected with both the train-pipe and the signal-pipe. Now, if the train-pipe pressure is raised one or two pounds above the normal ninety pounds, it causes the device to open a vent connecting the signal-pipe with the atmosphere, thus making the whistle blow in the cab. The engineer is thus informed that all the pipe connections in the train-pipe and the signal-pipe have been properly made and that a free and unobstructed connection exists in both. It is customary, of course, to test both the train-pipe and the signal-pipe of a passenger train immediately after coupling the engine to see that the trainmen have connected all the hose and opened the angle-cocks. But, while this test gives satisfactory assurance of the properly working condition of the brake and signal apparatus at the beginning of a run, it does not provide the engineer with a means of finding out whether such a condition is maintained. The closing of an angle-cock by a flying stone from the roadbed is an accident not unknown, and under ordinary conditions of service it might escape the knowledge of the engineer until the time came for a stop. If the angle-cock should happen to be closed near the locomotive the result would be that only a small percentage of the braking power would be available, thus presenting the elements for a disaster in case a quick stop was imperative.

AUTOMOBILE TURNTABLE.

Those who have to do with automobiles in garages and factories will be interested in a turntable that facilitates the handling of the cars and overcomes the unsatisfactory way of maneuvering by hand. The device shown by the illustration is manufactured by the Link-Belt Engineering Co., Philadelphia, and is essentially designed for easy operation. It con-



An Automobile Turntable.

sists of a cast-iron table fitted at its center with ball bearings and supported at the outer edge upon rollers. The latter are set in a concrete pit having a center pier upon which the ball-race is securely fastened. The wall of the pit is protected by an iron ring, or curb, which effectually prevents mutilation of the concrete edge. A brake is furnished and is depressed or let into the floor and top of turntable, insuring freedom from obstruction.

THE VALUE OF TAIL RODS.

A suburban railway company has as a part of its equipment two 1,000-horse-power cross-compound condensing Corliss engines, the engines being alike excepting that one has tail-rods. The engine with the tail-rods has been the more economical of the two, and has cost practically nothing for maintenance during the two years it has been running. A saving in steam consumption has also been one of the features of this engine with the tail-rods, and it can be run at about 10 per cent greater capacity than the other. In the three years which the engine without the tail-rods has run, it has worn out two sets of bull rings and the low-pressure cylinder has been worn down about 1 1-16 inch.—*American Electrician.*

LETTERS UPON PRACTICAL SUBJECTS.

A WARNING.

Editor MACHINERY:

Fred was a good, practical machinist, but he was of that class which works by rule-of-thumb methods and never takes the trouble of thinking about their work unless necessary. One day the foreman told him to stay over and reseat a 6-inch valve which was on the main steam pipe. As soon as the whistle blew Fred rushed round to the boiler house and carefully shut down the main steam valve; then he sent Will, the laborer, for the reseating tools while he rigged up the platform. Will soon returned and, as the platform was ready, they took out the screws which hold the top part of the valve on; Will gave it a wrench to pull it off, but it would not come, so Fred picked up a hand hammer and gave it a sharp blow. Instantly the top part flew off, catching Will a glancing blow on the leg and with a tremendous hiss a large volume of steam blew out, severely scalding both Fred and his laborer.

This almost fatal accident would have been avoided if the machinist had just thought for a moment about what he was doing when he shut down the main valve; that, although he had shut off the steam supply, yet the main pipe was still full of high-pressure steam which would take some time to condense. He should, of course, have opened the drip valve and blown it out before loosening the valve top. This is just one instance, but how many accidents, breakdowns, and wrecks can be traced to the similar cause—working with the hands only!

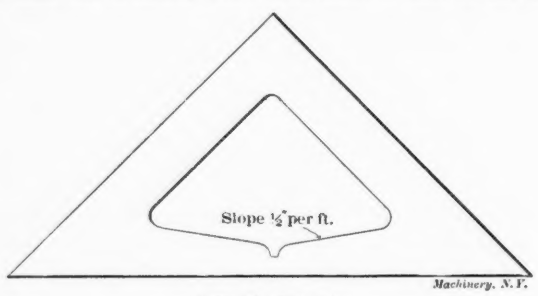
GEO. P. PEARCE.

Fort Hamilton, N. Y.

TRIANGLE FOR DRAWING I-BEAM SECTIONS.

Editor MACHINERY:

The sketch shows an alteration to a triangle which makes the drawing of the sections of I-beams and channels much easier than the ordinary way. The slant is that of the flanges



A Handy Triangle.

of the standard rolled sections, *i. e.*, 16 2-3 per cent or 2 inches per foot. This may be of service to those draftsmen who have some structural work to do, but not enough to warrant the purchase of a special triangle.

ROGER DEL. FRENCH.

Worcester, Mass.

KEEPING CUTTING TOOLS SHARP.

Editor MACHINERY:

The extreme importance of having cutting tools kept sharp is now generally understood. Like many other questions in shop practice, the truth of which are known, it is not usually capable of being actually demonstrated. The accompanying diagram shows the turning moment for three different 1½-inch drills each at three speeds, but at one feed. It tabulates the result of a few out of many experiments made under Dr. Nickolson, by a machine testing class of which the writer was a member. In the case of experiments marked *A*, through the edge chipping, the drill was sharpened before starting the third hole, and the results lie on a straight line. In the other cases the drills did the three holes without any grinding between the tests. In both cases the turning moment for the last hole was high above the probable curve. It should be noted that after all experiments the edge of the drill was not, so far as could be seen by the naked eye, the least dulled. This deterioration was not confined to any make or kind of drill, *B* being of English high speed steel and *C* a low carbon drill of American make. The work of the drills, drilling through about 4 inches of medium steel at .01-inch per revolution, was not by any means severe. In fact, judging by the amount of work done and the appearance of the drills, where the grinding was left to the option of the workmen, in the majority of cases they would not have been sharpened.

It may seem as if the slight increase in moment required is not important. Judged simply as a loss of power, it could be

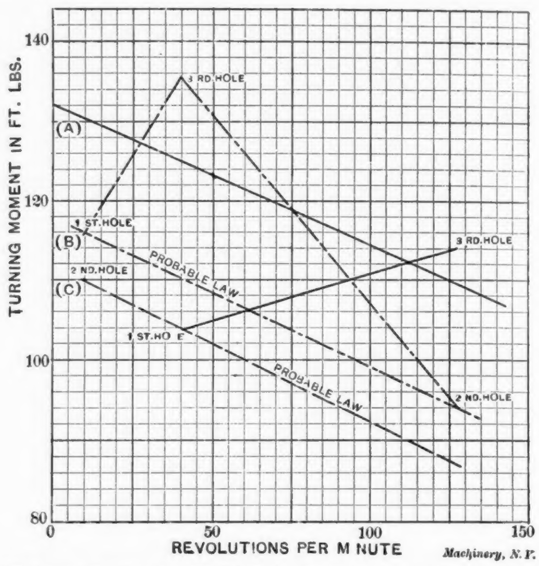


Diagram showing the Effect that a Dull Cutting Edge has on the Action of the Twist Drill.

neglected. But all the extra power goes toward heating the work and the tool, and produces many undesirable results. It would seem from the foregoing that tools should be ground after doing a certain amount of work whether the appearance suggests it or not. This is in accordance with the best practice to-day.

H. T. MILLAR.

Manchester England.

THE PROPORTIONS OF HAND TAPS.

Editor MACHINERY:

When using hand taps it has occurred to me that hand taps in sets of three are not made by manufacturers in such a manner as to give fair portions of work for each tap to do. All the taps I have ever been able to secure have been made in the same manner, *i. e.*, by giving a different length of chamfer to the different taps in a set; otherwise all three taps have had the same diameter in the angle of the thread and the same outside diameter. In my case, having had machine steel strips

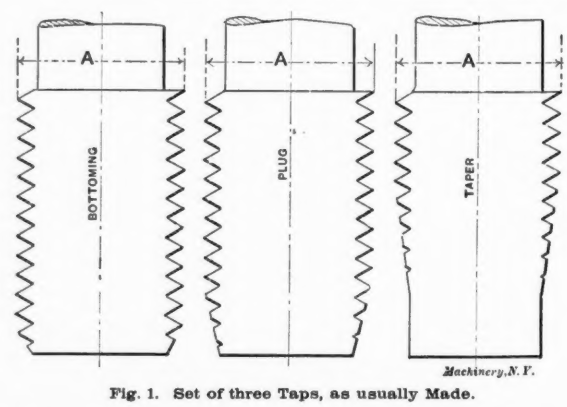


Fig. 1. Set of three Taps, as usually Made.

to tap holes straight through, I have found that the second and third tap had no work to do whatsoever, and as a consequence I have discontinued buying them. Now, the difficulty has been that the first tap (taper), having to do all the work for the tapping out of the hole, is put to altogether too severe a strain, and I have had much trouble about the taps breaking. As this became rather expensive and as I had an idea that I could overcome the difficulty, I made up a set of three

taps so proportioned in regard to outside diameter as to give to each tap a fair amount of the work to do. This set of taps I have now used with entire satisfaction for a time during which I ordinarily would have had to buy and break at least a dozen regular taper taps, and as these taps are still as good as new there is no telling how long they are going to last.

No. of Threads per inch.	V-THREAD.			U. S. S. THREAD.		
	Depth of Thread.	$\frac{4}{3}$ of Depth.	$\frac{1}{2}$ of Depth.	Depth of Thread.	$\frac{4}{3}$ of Depth.	$\frac{1}{3}$ of Depth.
3	.28868	.38490	.14434	.21651	.28868	.07217
3½	.24744	.32992	.12372	.18558	.24744	.06186
4	.21651	.28868	.10825	.16238	.21648	.05412
4½	.19245	.25660	.09622	.14434	.19244	.04811
5	.17321	.23092	.08660	.12990	.17320	.04330
5½	.15746	.20992	.07873	.11809	.15744	.03936
6	.14434	.19244	.07217	.10825	.14432	.03608
7	.12372	.16496	.06186	.09279	.12372	.03093
8	.10825	.14432	.05412	.08119	.10824	.02706
9	.09623	.12828	.04811	.07217	.09622	.02405
10	.08660	.11544	.0433	.06495	.08660	.02165
11	.07873	.10504	.03936	.05905	.07872	.01968
11½	.07531	.10040	.03765	.05648	.07528	.01882
12	.07217	.09620	.03608	.05413	.07216	.01804
13	.06662	.08880	.03331	.04996	.06660	.01665
14	.06186	.08248	.03093	.04639	.06184	.01546
16	.05413	.07216	.02706	.04059	.05412	.01353
18	.04811	.06412	.02405	.03608	.04811	.01202
20	.04330	.0577	.02165	.03248	.04328	.01082
22	.03936	.05248	.01968	.02952	.03936	.00984
24	.03608	.04808	.01804	.02706	.03608	.00902
26	.03331	.04440	.01665	.02498	.03328	.00832
27	.03208	.04276	.01604	.02406	.03208	.00802
28	.03093	.04124	.01546	.02320	.03092	.00773
30	.02887	.03848	.01443	.02165	.02884	.00721
32	.02706	.03608	.01353	.02030	.02704	.00676
34	.02547	.03396	.01273	.01910	.02544	.00636
36	.02406	.03208	.01203	.01804	.02404	.00601
38	.02279	.03036	.01139	.01709	.02276	.00569
40	.02165	.02884	.01082	.01624	.02164	.00541
42	.02062	.02748	.01031	.01546	.02060	.00515
44	.01968	.02624	.00984	.01476	.01968	.00492
46	.01883	.02508	.00941	.01412	.01880	.00470
48	.01804	.02404	.00902	.01353	.01804	.00451
50	.01732	.02308	.00866	.01299	.01732	.00433
52	.01665	.0222	.00832	.01249	.01664	.00416
54	.01604	.02136	.00802	.01203	.01604	.00401
56	.01546	.02060	.00773	.01160	.01544	.00386
58	.01493	.01988	.00746	.01120	.01492	.00373
60	.01443	.01924	.00721	.01083	.01444	.00361

Having had such success with my taps I have laid out a diagram and figured a table giving how much less in outside diameter the first and second tap should be made than the finishing tap in order to secure a set of taps giving good results.

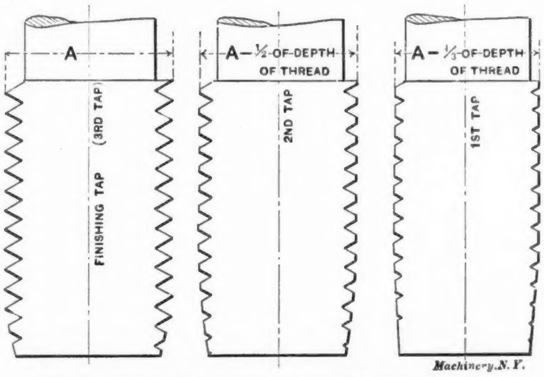


Fig. 2. Proposed Arrangement for Set of three Taps.

If we now first consider the diagram of the 60-degree V-thread, it will be noticed that the outside diameter of the first tap will equal the outside diameter of the finishing tap minus 4-3 of the depth of the thread; the outside diameter of the second tap will equal the outside diameter of the finishing tap minus ½ of the depth of the thread. It may occur to some one that the first tap is more than necessarily reduced in diameter, and that this tap has not as much stock to remove as the second tap, but considering the shape of the thread, it can easily be calculated that the area *A B C D* is larger than the area *C D E F;* in fact, the former area (*A B C D*) is in proportion to the latter (*C D E F*) as 9 to 6. It is obvious that the first tap ought to remove more stock than the second, because the top of the thread in the first tap has a smaller or less leverage than the top of the thread in the second tap. The finishing tap again, according to the diagram, only removes 1-16 of the whole area (*A B G*) of the thread. The reason for apportioning such a small amount to the finishing tap is to secure a nicely finished threaded hole.

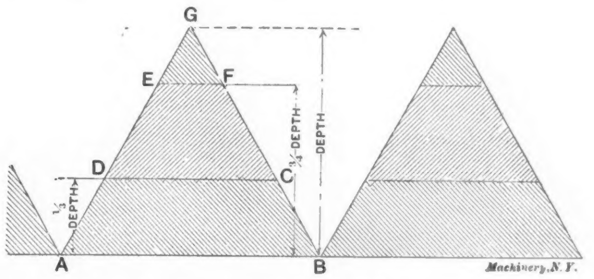


Fig. 3. Diagram showing Depth of Cut for V-thread.

Although I did not in the first place make my taps with any difference in the diameter of the angle of the thread, I would suggest that the first and second taps be made 0.001 or 0.002 inch smaller in the angle of the thread than the finishing tap. This will leave a very slight amount of stock to be removed by the finishing tap all over the thread, thus giving a nicer and smoother finished threaded hole than could otherwise be obtained.

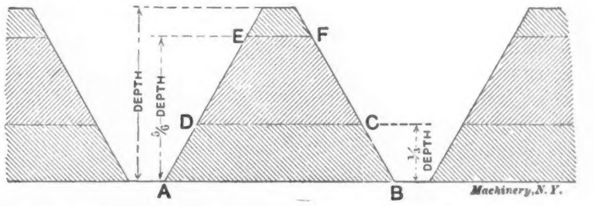


Fig. 4. Diagram showing Depth of Cut for U. S. Standard Thread.

If we now again consider the U. S. standard thread it will be found that the outside diameter of the first tap will equal the outside diameter of the finishing tap minus 4-3 of the depth of the thread; the outside diameter of the second tap will equal the outside diameter of the finishing tap minus 1-3 of the depth of the thread. The areas *A B C D* and *C D E F* will in this case be in the same proportion to one another as 31 is to 21 (10 to 7 approximately). A. A.

HINTS ON ESTIMATING THE COST OF WORK.

Editor MACHINERY:

It is very convenient for the estimator to have a list of the speeds and feeds of the machines in his shop. By means of this he can find the approximate time required for a given operation. Take, for instance, a drilling job. If we assume a proper speed and feed for the size of drill to be used in the given material, it is easy to calculate the time required to drill one hole. Of course the time required to "rig up" for the job, and the time used in removing the drill and starting a new hole, must be estimated from previous jobs.

In drilling some cast steel flanges containing two rows of 1 5-16-inch holes, 44 in each row, a Novo twist drill running 80 revolutions per minute with 1-64-inch feed was used in a universal radial drilling machine; a jig was provided to locate the holes. At this speed a drill would go through 1¼ inch per minute. The flanges being 1¾ inch thick, the drill would go through the flange in about 1½ minute. On a radial drill with a jig, the drill can be removed and started in a new hole in one minute, which, added to the time required for actual drilling, makes 2½ minutes on each hole. Hence about 22 holes per hour is about as good as can be done in actual practice under the conditions named.

In drilling a brass plate such as a condenser tubesheet, the drill can generally be run as fast as the machine will go. In a job of this kind 600 ⅝-inch holes can be drilled through 2 inches in 10 hours.

With modern high-speed steel for lathe tools, the limit is

usually with the work, as to speed, and, of course, a long slender shaft will not stand as fast speed and feed as a short and stiff one. The sketch shows an axle for an electric car which I recently turned up. For stock I was furnished with a bar of 5¼-inch steel sawn from a long bar with a cold saw.

Centering, squaring ends and taking off chuck and back rest occupied about two hours.

I found the piece to "run out" 3-16 inch in the middle, so one roughing cut would not leave this part running true. Using a Novo tool and running 70 R. P. M. with a feed of 18 turns per inch, the tool turned 1 foot in 4 minutes; grinding tool, etc., made the time for first chip ½ hour. For the second cut the speed was 80 R. P. M., and with same feed 1 foot was turned in 2½ to 3 minutes. Both cuts occupied about one hour.

Roughing out ends with the same speed (80 R. P. M.) took ½ hour each, and water finishing them at 36 R. P. M. took 1 hour each.

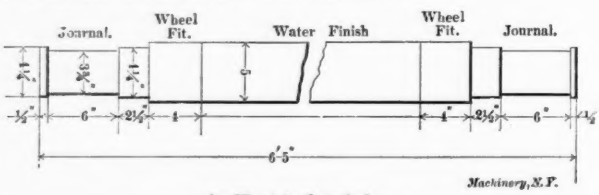
I left the wheel fits rough, and water finished the middle at 16 R. P. M. or 1 foot in 15 minutes or 1 hour for 4 feet. A shorter shaft could have been run faster, but this being so long and slender would have chattered and left the work rough.

The whole time for turning axle was:

Centering and squaring ends	2 hours
Roughing out (2 cuts)	1 hour
Roughing out ends at ½ hour	1 hour
Finishing journals	2 hours
Finishing middle	1 hour
Total	7 hours

This is not a "record breaker," but about the average time for a job of this kind, and a safe basis for the estimator to figure from.

A shaft with less stock to be removed would ordinarily have to be straightened, and no rule can be given for this, but it is safe to allow an hour for this operation.



An Electric Car Axle.

It is a good plan for the estimator, or for the ambitious mechanic who wishes to be promoted to that position, to keep a record of his time on various jobs and the time of others on important work if possible. This with a rough sketch will be of great value to him in "figuring on work."

One good point about the "piece work" system, so called, was that it trained mechanics to note the cost of doing work. It is not a safe rule that if we know the time for a given piece of work to assume that a piece twice as large will require twice the time. Neither is it safe to surmise that it will take twice as long to machine two pieces as it would one. Once I had twenty-five shafts to turn. It took me eight hours to turn the first and after that I turned three a day.

The time required to do a job in many cases depends on "the man behind the lathe," so that it is not safe to estimate from the most rapid workman's time, nor is it safe to use the other extreme, but a fair average must be found.

Portland, Me. H. K. GRIGGS.

REVERSING LEVER PROBLEM.

Editor MACHINERY:

The accompanying sketch shows a practical reversing lever problem diagram. Its solution requires considerable time as well as a knowledge of mathematics. Having occasion recently to solve the problem I reduced it to its simplest form—an arithmetical formula—and made a memoranda of it for future use. The thought occurred to me that it would probably be serviceable to many of your readers also.

We have given, the distance, c, between two center lines of motion one which is of a length, a, and the other of a length, b. The center of the radii R and r lies in the perpendicular bisector, which represents the lever in mid-position, and this point, of course, is the fulcrum of the lever. Then the distances x and y on the perpendicular bisector of the horizontal center lines of motion are derived thus:

$$x = \frac{a\,c}{a+b}$$

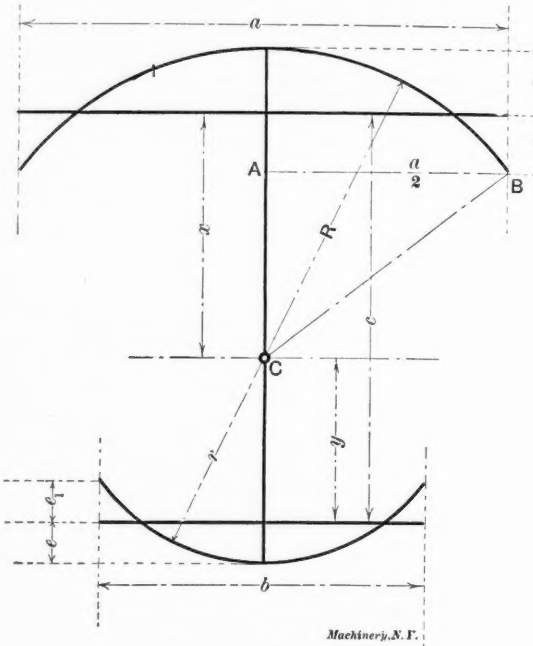
$$y = \frac{b\,c}{a+b}$$

From which R is found by formula:

$$R = \frac{a^2}{16\,x} + x, \text{ and}$$

$$r = \frac{b^2}{16\,y} + y,$$

which are the proper dimensions for a theoretically correct reversing lever; that is, one which vibrates equally each side



A Reversing Lever Problem.

of the center line and the arc of which passes as far above the center line of motion as it does below it, or so that $d = d_1$ and $e = e_1$. FRANKLIN H. SMITH.

Buffalo, N. Y.

Mr. Smith did not give the derivation of the formula in his letter, but as same will doubtless be of interest to those desiring to use it we append it herewith. It is apparent from the conditions of the problem and the diagram that $R - d_1 = x$; also $d = d_1$. Drawing the radius $B\,C$ and $A\,B$ parallel to a we have a right triangle in which $\overline{A\,C}^2 = \overline{B\,C}^2 - \overline{A\,B}^2$. But $B\,C = R$, and $A\,B = \frac{a}{2}$. Hence

$$\sqrt{R^2 - \frac{a^2}{4}} + d = x. \text{ Adding both values of } x$$

$$R + \sqrt{R^2 - \frac{a^2}{4}} = 2\,x. \text{ Simplifying}$$

$$R = \frac{a^2}{16\,x} + x$$

In the same way it can be shown that

$$r = \frac{b^2}{16\,y} - y.$$

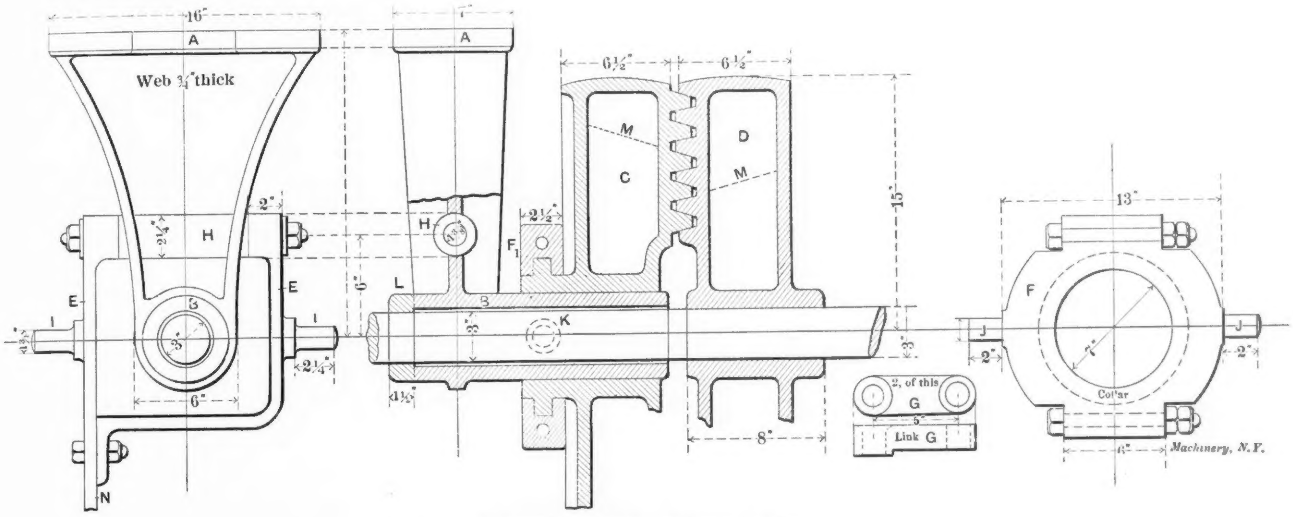
A COMBINED CLUTCH AND LOOSE PULLEY.

Editor MACHINERY:

The type of loose pulley shown in the sketch has long been known as the best type for use in planing mills, or other woodworking establishments, where the machines are run at a high velocity. The tight pulley is placed on the first mover or lineshaft, with the loose pulley placed beside it, but instead of the loose pulley running on the shaft it runs on a

sleeve cast on the hanger. The sleeve is bored large enough for the shaft to pass through without touching it, being, say, ⅛ inch larger. Neither a collar nor a shoulder is required on the sleeve in the combined loose pulley and clutch, as the clutch collar and lever keep the loose pulley in place. When the belt is on the loose pulley, the machine, the belt, and the loose pulley are all idle. Consequently there is no wear, and no attendance is required.

To start a machine when the clutch is omitted, take hold of the belt and put it in motion, which is easily done in the case of woodworking tools when the feed is thrown off and the machine not cutting. The belt being thus put in motion, it is easily moved on to the tight pulley by the belt shifter which is of the ordinary type, consisting of a sliding bar with verticals on each side of the belt on the side of the pulley toward which the belt runs. To stop the machine shift the belt on to the loose pulley by the belt shifter and the entire system—machine, countershaft and loose pulley and belt—is idle and all wear stops. The loose pulley runs only while the belt is being shifted. It will readily be seen that such a pulley needs but little oil, say a few drops once a month, and if the device mentioned by "Bristol" in the May issue of Machinery were used in the bore of the pulley it would never need oil.



A Combined Clutch and Loose Pulley.

This device is admirable where there are several lines of shafting all driven from the main line or first mover; any one or any number may be idle and the rest of the mill running for any length of time, and no wear takes place on any of the loose pulleys. It may be used in the driving of other machinery, which requires the full power to put it into motion, but in order to do this it would require the combined clutch and loose pulley like the device shown in the drawing which consists of a hanger and sleeve *A*, the latter bored to let the line shaft through as heretofore mentioned and turned to fit the loose pulley easily. The part of the hanger which is bolted to the ceiling or floor should be planed square with the sleeve. *C* is loose pulley with clutch grooves on one side. *D* is a tight pulley, also made with clutch grooves on one side, and is fastened to the lineshaft *K*. *E* is a wrought iron lever for throwing the clutch in and out; the fulcrum pin passes through the boss *H* on the hanger. *F* is a loose collar which fits on the collar turned on the hub of the loose pulley *C*, and is for throwing the clutch in and out. *G* is one of two links connecting pins *I* on the lever and *J* on the collar. The dotted circle *K* shows the position of the pins when the collar is in place; this collar and its pins are of cast steel. *L* is a collar secured to the shaft by conical point setscrews to keep the hanger from springing out of place. The dotted lines *M M* show position of small oil pipes for oiling clutch faces, which is required occasionally to prevent sticking.

The belt being on the loose pulley, the machine is started by throwing in the clutch by the lever *N*. The belt is then shifted onto the tight pulley, and the clutch faces moved apart to prevent wear. To stop, simply shift the belt as previously described. A separate starting lever and a belt-shifter are, of course, required.

So thoroughly do I believe in this type of loose pulley because of my own experience with it, I would not fear to build and erect them (if I had a foundry and machine shop) under a warranty against all natural wear for ten years.

Syracuse, N. Y. C. E. Mink.

ESTIMATING THE COST OF MACHINE WORK.

Editor Machinery:

A matter to be observed when estimating for a job is to have the estimation in such form as is easily compared with the cost account when the work is completed. While in Mr. Webster's method of tabulating—described in the July issue—the operation and quality of labor to be used on each seem very good, his method of adding the shop charges might be improved. By shop charges I mean the items of shop superintendence and part of the 33 1-3 per cent added to the account at the end.

The cost of an operation which involves the use of a machine should be estimated with regard to the interest and depreciation on the outlay. A $1,500 machine even when worked by cheap labor is perhaps as expensive per hour as a toolmaker working without a machine. It is quite conceivable, indeed it is very probable that some operations need more supervision than do others. Therefore it is natural that each operation should bear its share of the shop charges in proportion to the amount of capital which, so to speak, is being used at the time and the relative amount of "non-productive labor" it entails. This can not be gotten exactly, but the general run of shop operation could be divided into seven or eight classes, each with its different shop rate. The only alteration in the given case would be another column in which the shop rate—which would generally be independent of the quality of the labor—would be entered. While on the subject of labor, does not this job need rather a high percentage of boys—65?

Manchester, England. H. E. Millar.

[The percentage of boys' labor in time is 71.5 per cent, and in cost 52 per cent.—Editor.]

TO CHANGE GEAR RATIOS WHEN THE NUMBER OF TEETH AND CENTER DISTANCE IS FIXED.

Editor Machinery:

A problem in gearing which was quite out of the ordinary came to my notice some time ago. Perhaps the problem and its solution may interest some of your readers. A machine had been built with two of its shafts a certain distance apart and each running with equal number of revolutions. In the course of time it became necessary to reduce the number of revolutions of one of the shafts. It was desired to use the same pitch gears and to keep the same center distance, so the facts were that we had a certain number of teeth to utilize and desired a certain number of revolutions of each of the two gears. Therefore the problem was to obtain the number of

teeth in each gear. This was obtained by dividing the known number of revolutions by the desired number of revolutions per minute, and adding one to this quotient. Then divide the whole number of teeth in both of the equal gears by the number obtained and the quotient will be the number of teeth in the driving gear. This number taken from total number of teeth will show the number of teeth required in the driven gear. Example: Assume a driving shaft running with 100 R. P. M. and that it is desired to revolve the driven shaft 60 R. P. M.; total number of teeth to be utilized is 120. $\frac{100}{60} = 1.66 + 1 = 2.66$. $\frac{120}{2.66} = 45 =$ number of teeth in driving gear. $120 - 45 = 75 =$ number of teeth in driven gear.

If it had been desired to increase the number of revolutions in the driven instead to decrease them the respective tooth numbers would have been obtained by dividing the required number of R. P. M. by the R. P. M. of the driving gear and adding one to this quotient. Then, dividing the total number of teeth to be utilized by the number already obtained, the resulting quotient would be the number of teeth in the driven gear. This number taken from total number of teeth will show the number of teeth required in the driving gear. Example: Assume a driving shaft running with 100 R. P. M. and that it is desired to revolve the driven shaft 150 R. P. M. Total number of teeth to be utilized is 120. $\frac{150}{100} = 1.5 + 1 = 2.5$. $\frac{120}{2.5} = 48 =$ number of teeth in driven gear. $120 - 48 = 72 =$ number of teeth in driving gear.

C. E. Josselyn.

Bridgeport, Conn.

A FEW ITEMS WHICH HAVE HELPED ME.

As I draw circles with the compasses, where the centers are not well defined I draw a little free hand circle around the center. Then I don't have to hunt for the centers when inking.

If I have to make a drawing on a scale of 3 inches to the foot, as often happens, and have no scale handy that reads to this proportion, as also often happens, I look at it as ¼ inch = 1 inch. That is, suppose I have a measurement of 7⅝ inches. I read it as 7 quarter inches which I count off on my rule and ⅝ of a quarter inch. As most scales are divided at least as close as 1/16 inch this latter will be half way between ⅛ inch and ¾ inch, which are represented by the ⅛-inch and 3/16-inch marks on the scale. If I have to draw a circle of a given diameter to quarter size I use the same means to get the radius except that I read of ⅛-inch divisions and fractions of eighths instead of quarters and thus get the radii direct. All of which is a lot easier, and more likely to be correct than figuring it out.

I have to do drafting at different times in a number of places. We all know that the usual size of instrument case is not convenient to carry in the pocket, for men are apt to have steady use for about all the pockets they are allowed. After trying for some time I ran across a salesman who was willing to make up a set to suit me, of Riffler instruments which takes up little room and fills the bill. It consists only of a 5½-inch compass with the usual pencil and pen and lengthening bar. These are slip-joint instruments so changes can be made instantly. If I want to draw circles of 1/16 inch or even less radius I can do them just as well as with a bow instrument for there is a hair-spring attachment. If I want to use a ruling pen the lengthening bar makes a nice handle for the pen. By changing the lead for a compass needle I have a hair-spring divider, though when I get rich I am going to get an extra pencil holder and keep a point in it for such use. A short scale completes the outfit.

When I am going where I have no drawing-board I can usually buy a molding board such as "mother used to use," which makes a good board when two edges are planed off at right angles. They cost only a few cents so if one warps you can buy another. It is surprising how much work can be done on small sheets if you only get used to it. As for having two edges at right angles I am still old-fashioned enough to do my penciling with the triangle but I like to ink all vertical lines with the T-square. It is a great time saver over fooling with a set-square.

A T-square with a detachable head fastened by a dove-tail piece on the blade is a handy thing for traveling to say nothing of its being easily trued up.

How do the rest of the boys feel about doing designing by the hour; not where you have a steady job but where you undertake a certain job? The quicker you think up a scheme and the fewer futile attempts you make the less you get. On the other hand very few people can be persuaded to agree to pay a price for a lump job that makes it safe for the designer. Why should a designer do his work on a different basis from a lawyer or a doctor who charges what he considers his services worth? There would be the same check on his charges as on theirs for unless he pleased his clients there would be few new ones coming.

Entropy.

GROUPING MACHINERY'S DATA SHEETS.

Editor Machinery:

The data sheets published with Machinery are a great aid in machine design. We have a number of sets of them in this drawing room, and in order to find the required information

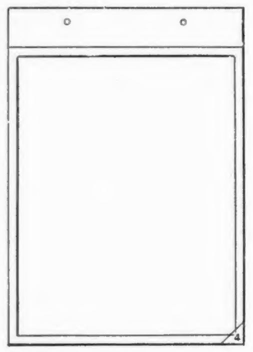


Fig. 1.

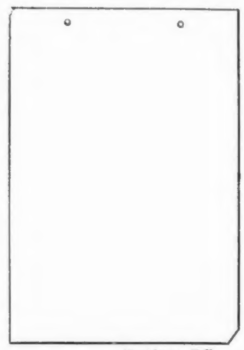


Fig. 2.

quickly I have arranged the sheets in groups in a binder according to an index of which I send you a copy herewith. On one sheet of each group, I mark the group number in the lower corner. On the other sheets I cut off the corner, Fig. 1. By

MACHINERY'S DATA SHEETS—INDEX.

1. Mathematical Tables.
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Size of Index Sheet 6 x 9 inches.

this arrangement I can at once find the required group of sheets. In order to get all the sheets uniform I use a cardboard template for punching the holes and trimming the corners as shown in Fig. 2.

K. B.

SHOP KINKS.

A DEPARTMENT OF PRACTICAL IDEAS FOR THE SHOP.
Contributions of kinks, devices and methods of doing work are solicited for this column. Write on one side of the paper only and send sketches when necessary.

TO PREVENT BABBITT FROM EXPLODING.

To prevent babbitt from exploding and to insure good bearings, put two or three teaspoonfuls of kerosene oil into the box before pouring the babbitt. H. K. GRIGGS.

Portland, Me.

TO REPAIR A FEED OR OTHER WATER PIPE TEMPORARILY.

To effect a temporary repair of a cracked water pipe mix a moderately stiff putty from red and white lead, with boiled linseed oil, and work into it some hemp chopped into short lengths; lay it over the crack in a moderately thick mass and wrap some strips of canvas tightly around the pipe overlapping both ends of the crack, and finish by sewing marline hard over the strips of canvas. JAMES A. PRATT.

Howard, R. I.

HOW TO SET A PLAIN DIE IN A PUNCH PRESS.

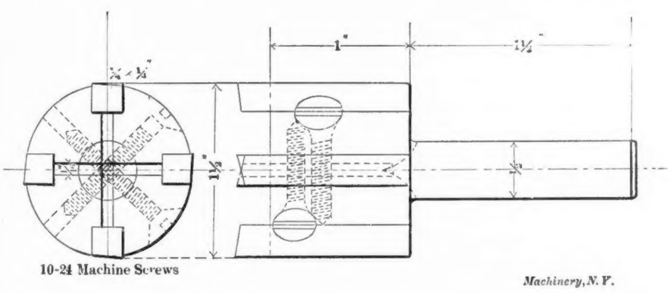
To set a plain die in a punch press adjust the male portion of the die so it will not go more than 11-32 inch below the surface of the female die; then twist the female die around to the right as far as it will go; mark it with reference to some stationary part of the press, and then turn it to the left, again marking in the same way; then turn the female die one-half the distance between the two marks, and now screw it down by tightening the screws alternately, being careful not to move it. In this way it is easy to get the space divided up accurately between the two parts of the die.

Winnetka, Ill. FRANK PAVLIK, JR.

"HURRY UP" END MILL.

Editor MACHINERY:

I herewith enclose sketch of a "hurry up" end-mill which I made recently, and it is certainly a winner. The body and shank is made from tool steel, one piece, and is slotted well back to the shank. The cutters are made of "Rex" steel, ground off on the sides to fit their seats in the body, and held by two 10-24 machine screws. The cutters may be ground by



hand on an emery wheel, then placed in position and the table raised, thus pushing the cutters up so the cutting ends are even. The binding screws are tightened as before, which, if done with a little care, adjusts the ends very close. This cutter, of course, was designed for the vertical milling attachment. L. E. MURRAY.

Syracuse, N. Y.

HARDENING A SCRATCH AWL.

After a great deal of experimenting I have discovered a most effective way to harden the points of scribers (or scratch awls as they are often called) so they will be hard and at the same time tough enough to "stand the racket." Here it is:

Heat the point of the scriber over an alcohol lamp or bunsen burner, leaving the extreme point out of the flame to avoid the danger of overheating. Hold a thin piece of ordinary soap in the hand over a cup of water, and when the scriber has reached a nice "cherry red," push it down through this soap into the water below. Draw the temper to a very dark straw. R. A. LACHMANN.

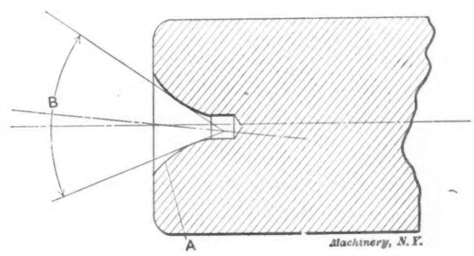
Chicago, Ill.

TO SUCCESSFULLY COLD WATER ANNEAL TOOL STEEL

To successfully "cold water anneal" tool steel, I find that the following method gives by far the best results of any of the various methods I have tried: The steel is heated and allowed to cool down in the usual way. Before being immersed in the water it is smeared with a bar of cheap alkali soap; it is then dipped into a pail of preferably luke-warm soap water, and quickly withdrawn, after which it is again smeared with the soap and dipped again and then allowed to cool. If a large piece is to be annealed, repeat the above until cool, this being necessary as a large piece retains its heat longer than does a small piece. Care should be taken not to get any soap or soap water in the tank containing the solution used for hardening. C. F. EMERSON.

CURVED SIDE COUNTERSINK IN MANDREL FOR TAPER LATHE WORK.

The accompanying sketch of the end of an arbor shows a way of countersinking which works much better than the standard angle of 60 degrees (shown by the line *B*) on taper work that is turned on a lathe without a taper attachment. The dotted lines show, exaggerated, the effect that throwing



over the tailstock has on the fit between the centers of the lathe and the arbor. The curved side countersink, of course, is not recommended for straight work, or for use on a lathe provided with a taper attachment. M. H. BALL.

Watervliet, N. Y.

TO START A THREADING DIE.

We all know how easy it is to tap a hole out of square but many of us don't realize that in running a die over a piece without the aid of a die holder made to fit in the tail stock spindle of a lathe, it is an easy matter to cut the thread out of line, as shown in Fig. 1.

FIG. 1

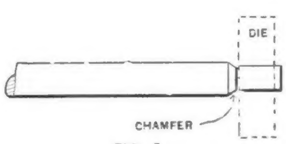


FIG. 3

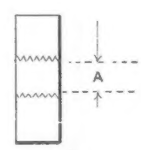


FIG. 2

Machinery, N.Y.

In order to eliminate this possibility I generally turn down the front end of piece to be threaded (leaving the piece sufficiently long so it can be cut off if desired) so that the die can just push over without shake; in other words, I turn the end to the root diameter of the die as shown in Fig. 2 at *A*. Fig. 3 shows how nicely the die starts true by using this method.

Chicago, Ill. R. A. LACHMANN.

* * *

An exchange says that in hard soldering or brazing with borax direct, difficulty is encountered from the great bubbles formed by the salt, which easily breaks away from the surfaces to be soldered or brazed. Another objection is that the parts must be carefully cleaned each time prior to applying the borax. It is advised that instead of using borax to use its component parts—boric acid and sodium carbonate. The heat acting on these causes them to combine in such a manner as to produce an excellent flux without the objectionable features of borax.

SHOP RECEIPTS AND FORMULAS.

This page will be used for the publication of shop receipts which the contributors know from experience to be practicable. Nearly all readers of MACHINERY can add something, and it is desired that they use this page as a medium for exchanging useful formulas, thus putting them on record for the general benefit of the craft. All published receipts are paid for.

TO ANNEAL STEEL OR IRON.

Smear the iron or steel with tallow, and heat slowly in a charcoal fire until it is a dark red. Allow it to cool itself. This method is all right for very hard tool steel.

Schenectady, N. Y. R. B. CASEY.

IRON CEMENT.

For plugging holes in castings a good cement is made from 80 parts of sifted cast iron turnings, 2 parts of powdered salammoniac, and 1 part sulphur, made into thick paste with water fresh for use. DAVID MELVILLE.

Detroit, Mich.

ATTACHING CLOTH TO IRON.

Heat the iron so it will be just too hot to touch with the bare hand, put on a coat of red shellac; have the cloth already cut, applying it quickly, and press firmly in place.

Howard, R. I. JAMES A. PRATT.

DRILLING COMPOUND.

A good drilling compound is made by adding 1 pound common soda to 4 quarts water, and 1 pint machine oil. Let stand for about one hour and it will be ready for use. This will not rust the machines and is clean to work with.

Winnetka, Ill. FRANK PAVLIK, JR.

RUST PREVENTATIVE.

To make a mixture that will prevent hardware and machinists' tools from rusting, take one-half pint of Demar white varnish, and mix it well with one gallon of turpentine; use as a wash. When the polished surfaces are thoroughly covered with a thin coat of the varnish it will show scarcely any, but will preserve the polish for years, if it is not scraped off with something very hard. H. E. WOOD.

Pearl River, N. Y.

DIE SINKERS' IMPRESSION WAX.

In the following I give two receipts for die-sinkers' impression wax. In the first the exact proportions of some of the ingredients are not given, but the maker can use his own judgment, gradually adding more of one than the other until the right consistency is obtained. 1. Beeswax, 6 parts; white wax, 1 part; a small quantity of cornstarch; sufficient Racine castor oil to make it of the desired consistency. Add stearine if too soft. 2. Another receipt is two parts of beeswax, and one part bayberry wax. I have found powdered chalk useful to remove stickiness of this wax. C. W. SHELLY.

Niagara Falls, N. Y.

BLUE PRINTING FORMULA.

I have used the following recipe for blue prints with much satisfaction. The same formula may be applied for postal cards on which it is desired to print landscapes or similar views. Make a solution as follows: Water, 3 ounces; ammonia citrate of iron, 300 grains; oxalate of potash, 75 grains. Dry in the dark, print and then develop in the following: Water, 3 ounces; nitrate of silver, 15 grains; citrate of soda, 150 grams. Add ammonia to dissolve the precipitate, and acetic acid until slightly acid. Wash slightly and dry. I have found this to make a better blueprint in every detail than any other of the various known recipes.

Orange, Conn. ROBERT B. OTIS.

LUBRICANT FOR PIPE SCREW THREADS.

The best "dope," so called in shop parlance, that I have ever seen used for making pipe connections, is composed of 1 pint of "black strap" machine oil, ½ pint graphite, ¼ pint of white lead, and a teaspoonful of flour emery. These proportions are not exact, but they are substantially what are used. The object of the flour emery is to polish the threads as they are being screwed together. The graphite, white lead and oil make a fine lubricating mixture, which has enough consistency to stop incipient leaks. I have seen many large pipe radiators made up using this mixture, and they never leaked a drop when the steam was turned on.

Altay, N. Y. M. E. CANEK.

CHEMICALS FOR BLUEPRINTS.

To make blue-print paper use citrate of iron and ammonia, 1⅞ ounces dissolved in 8 ounces of water, and red prussiate of potash, 1¼ ounces dissolved also in 8 ounces of water. Keep in separate bottles until wanted for use. When wanted for use, measure equal quantities from each of the above bottles. Shake so as to mix it well. It is then ready for putting on the paper. When the two are poured together, the mixture must be kept away from white light and should be applied in a room illuminated with a ruby light only. The paper must be dried in this room and kept in the dark until used. One ounce of mixed chemical will cover about 4 square feet of paper. DAVID MELVILLE.

Detroit, Mich.

PAINT FOR FITTING AND SCRAPING.

To make a paint for fitting and scraping get five or ten cents worth of scarlet vermilion (powder) at any store where paint is sold. Melt a tablespoonful of lard and mix into the dry paint until like thick cream, and when cold is just right. The vermilion is very fine and has no grit in it so that the least touch of the mixture shows.

This is better than the tube paint generally used, as being mixed with animal oil, it will stand exposure to the air for a year or more without drying; but the tube paint is mixed with vegetable oil and will soon harden on exposure to the air. Any colored paint powder can be used, which is preferred. To test for grit take some between the thumb and forefinger.

F. W. B.

* * *

MARKING PAINT.

In shops making a business of repairing machinery, it is generally necessary to mark the parts of machines in some way so that they may be properly reassembled. This is especially true in railway shops, where the marking is necessary more for the purpose of distinguishing the parts of different engines. The best way to mark such parts, of course, is to stamp them with steel dies; but this is not always practicable, and, in the absence of such means of marking, it is customary to use a marking paint made of white lead mixed with turpentine to a thin consistency. Such paint dries quickly and when dry is not easily removed. It has the advantage of showing up fairly well on greasy surfaces; but it is better that the surfaces to be marked should be well cleaned with kerosene oil before marking. F. EMERSON.

Newark, N. J.

* * *

In this age, when iron and steel are so plentiful and such a necessary part of our civilization, it is difficult to realize that only a few hundred years ago iron was so scarce that it was ranked with the precious metals. Even as late as the time of Edward III. of England, iron was so rare that the pots, spits and frying pans of the royal kitchen were classed among the king's jewels!

* * *

An interesting improvement in the process of obtaining a high vacuum is reported to have been developed in England by Dewar. The method commonly employed for obtaining the high vacuum required in incandescent lamps and similar apparatus is the quick-silver process, *i. e.*, Sprengel's air-pump. This method, while fairly satisfactory, is slow in operation and requires a great volume of apparatus where a large number of lamps are being manufactured. The Dewar process is very simple and quick in operation. It depends upon the absorption power of charcoal when cooled. It appears that as charcoal is cooled down to the temperature of liquid air the absorption of air takes place so energetically that, if the charcoal is contained in a closed vessel, the air in the latter is soon exhausted. Charcoal made from cocoanut shells is preferably used and this is contained in a tube attached to the vessel to be exhausted. The tube is immersed in liquid air and in a few minutes the air in the tube and vessel has been absorbed. The method also possesses the advantage that the moisture is condensed in the tube.

HOW AND WHY.

A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

Give all details and name and address. The latter are for our own convenience and will not be published.

1.—R. B. R. Kindly give me an explanation of the process of coating gray iron castings with tin, or a composition, such as is used in the covering of castings employed in the manufacture of meat choppers and similar machines.

2.—G. W. E. Kindly explain through your columns how to magnetize U bends of steel having positive and negative poles at the ends of the U. The steel is 5/16 inch by 2 inches, and the arms of the U are 6 inches long bent the flat way. I desire to use magnets for the fields of an electro-magnetic generator, or "Magneto." I have tried to magnetize them on the fields of a 500-volt Westinghouse generator (old style); and I also have tried winding No. 21 B. & S. gage magnet wire on each end of the U, having one pound in each coil and running a current through them from a direct-current 110-volt machine with 6-16 candle power. Neither method, however, has given satisfactory results, though the steel was as hard as heating to a cherry-red and dipping would make it. Is there any special brand of steel better to use than the ordinary spring steel?

Answered by Wm. Baxter, Jr.

A. The best way to magnetize permanent magnets is with wire coils, in precisely the way you tried. You did not meet with success principally because you did not pass enough current through the coils. The quality of the steel may also have had something to do with it. The best steel for permanent magnets is the old-time tool steel. Make a keeper of soft iron, about 1 inch square, and fit it well to the ends of the magnet; then put the wire coils in place, connect them in series, and while the keeper is against the ends, connect the coils in a 110-volt circuit. Let the current pass for a few seconds, or until the coils begin to warm up, disconnect, and the magnet will be well magnetized. The magnet should be hardened in salt water with ice in it, and after being polished should be drawn until the faintest shade of straw color appears. After this magnetize.

* * *

DANGER OF COCKS IN STEAM GAGE PIPES.

It develops that the disastrous explosion which occurred on the gunboat *Bennington*, July 21st, was indirectly caused by a cock being closed between one of the boilers and its steam gage. It appears that the fireman was instructed to shut off the air cock which is provided for discharging the free air in a boiler when firing up, but by mistake he closed the steam gage valve. Two boilers, A and B, were being fired up, and although a pressure of 135 pounds was registered on the steam gage of boiler A, that of boiler B failed to register more than 5 pounds pressure. Notwithstanding this the fireman kept working the fires and shoveling in coal until the disaster occurred. It also developed in the naval inquiry that the safety valves were not regularly lifted from their seats, and the sentinel valves had not been overhauled for over a year. The hand gear for lifting the safety-valves was not in working order, and there is no record nor direct evidence that the safety valves had been tested.

While there is undoubtedly good evidence of gross neglect on the part of officers in charge, there is also evidence of a serious mistake in boiler design in putting a cock in the pipe leading to the steam gage. While we are not conversant with all the conditions that surround a boiler in naval service, we know that with other steam boilers this practice has been found very unsafe and is almost universally condemned. Of course it is very convenient to have a cock so located, as the steam gage can then be removed at any time and tested; but such construction is so fraught with the possibility of disaster that we believe that it should be strictly prohibited. It is better, a thousand times, that steam be blown off a boiler so that the steam gage may be tested or changed, than to run the risk of having a disaster occur because of a simple convenience. Of course, a steam gage is not a direct safety appliance except as it appeals to the sense of the fireman; but safety valves sometimes stick even when attended to with great regularity. In boiler practice we should neglect no means making for security, and if it is proven that a cock in a steam gage is dangerous, by all means abolish it.

A case illustrating the danger of a cock in a steam gage pipe occurred in the writer's experience: A locomotive had been overhauled, during which work the safety-valves had been ground and the springs reset by guess, as was the custom. The boiler was being fired up in the yard but the fireman, who was not over-well blessed with common sense, finally complained that he was not able to get the necessary pressure. The best he could get was about 35 pounds. The man who was responsible for setting the safety-valves happened to go out to examine some other matters when he discovered that the cock in the steam gage pipe was closed. Upon opening it the pointer at once went around the circumference of a 250-pound gage and struck the stop pin. Needless to say the pressure was quickly reduced by relieving the springs. There is little doubt that had it not happened that this timely discovery was made a disastrous explosion would have occurred.

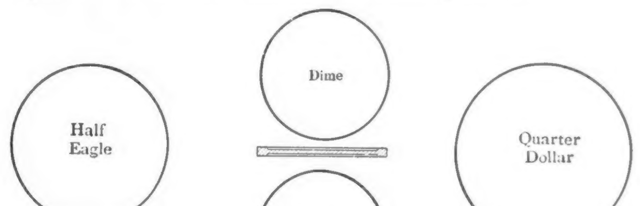
* * *

PRESSURE REQUIRED FOR STAMPING UNITED STATES COINS.

The following table and illustration are from a pamphlet issued by the E. W. Bliss Co., Brooklyn, N. Y., describing their line of minting machinery. The pressures quoted in the table are from tests made at the United States Mint, Philadelphia.

U. S. Coin.	Metal.	Tons Pressure.	U. S. Coin.	Metal.	Tons Pressure.
Double Eagle	Gold	155	Half-dollar	Silver	98
Eagle	Gold	110	Quarter-dollar	Silver	60
Half eagle	Gold	60	Dime	Silver	25
Quarter-eagle	Gold	35	5 cent nickle	Nickel	60
Standard dollar	Silver	160	1 cent	Copper	40

The above pressures are correct within about 5 per cent.



These Outlines show the Diameter and Thickness of the U. S. Coins mentioned in above List.

* * *

What is probably the most extensive dust collecting system on this continent is now in successful operation in the planing mill and cabinet shop of the Canadian Pacific Railway at its Angus Shops in Montreal.

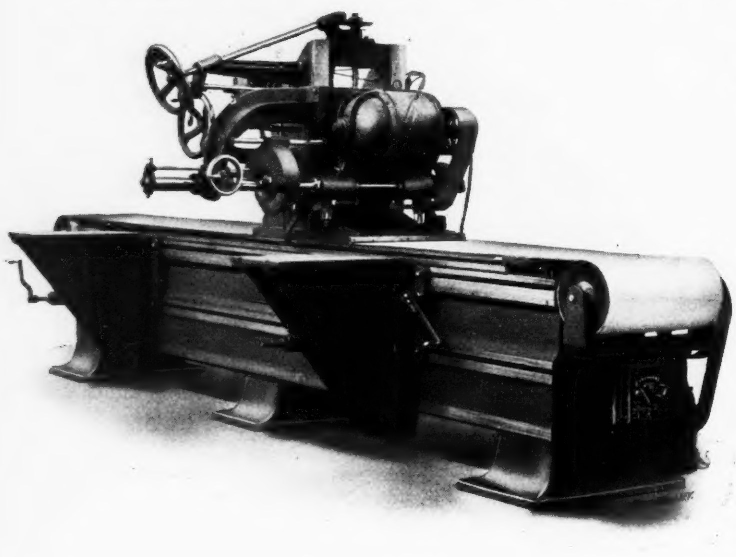
MACHINERY AND TOOLS.

A MONTHLY RECORD OF NEW APPLIANCES FOR THE SHOP.

TRAVERSE GRINDER.

This machine has been designed especially for grinding manganese cast steel centres of railroad frogs and crossings, locomotive cross head guides, finishing butt ends of connecting rods, etc., and is, also, adaptable for grinding the ways between the V's on lathe beds, and for other grinding purposes.

The bed is cast in box form and is braced internally by means of cross girts and can be furnished in varying lengths, the illustration showing one 15 feet long, which permits a 12 foot longitudinal travel of the wheel. The saddle which carries the wheel is traversed by a 2 H. P. motor through a rack



Traverse Grinder.

and pinion at a speed of 15 feet per minute, and is automatically reversed at each end of the stroke which may be adjusted to any length up to the limit of the machine.

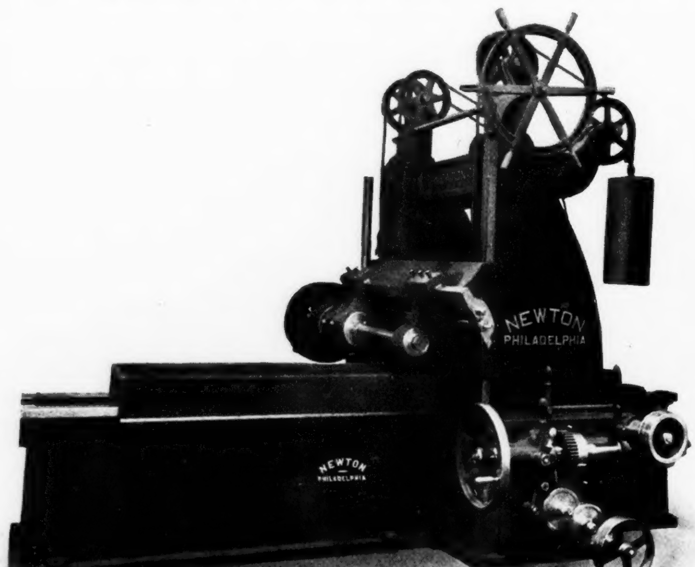
The grinding wheel, driven by a 5 H. P. motor at a speed of 1,500 R. P. M., has a horizontal movement in the direction of its axis of 15 inches, and a vertical movement of 11½ inches, these movements being operated, one by the hand wheel attached to the bracket supporting the grinding wheel shaft, and the other by the large hand wheel whose shaft is set at an angle. The third hand wheel is for moving the saddle by hand when desired. The motor is adjustable on the saddle, for the purpose of tightening the belt driving the grinding wheel shaft.

The tables are horizontally adjustable along the bed, by means of a screw, operated by the socket wrenches attached to them. A pump and system of piping is provided for supplying water to the grinding wheel, and canvas curtains are used to protect the sliding ways on the top of the bed from flying grit.

All controls are located in the saddle; a lever, not visible in the cut, being provided for reversing the saddle by hand when necessary. This grinder is built by the Cincinnati Shaper Co., Cincinnati, O.

NEWTON SLAB MILLING MACHINE.

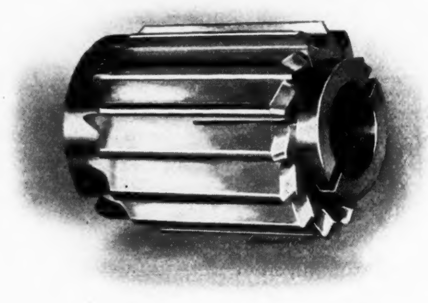
The accompanying half-tone shows a plain slab milling machine recently furnished to one of the large electric companies by the Newton Machine Tool Works, Philadelphia, Pa. The spindle of the machine is 5 inches in diameter, and will swing a cutter up to 12 inches in diameter; the spindle has an adjustment on rail of 6 inches for convenience in setting cutters. It is driven by a 7½ H. P. motor through gearing by hardened steel worm and phosphor bronze worm wheel of steep pitch. The cross rail is counterweighted, and is of new design with inclined face. With this design of rail, the thrust of cut is transferred directly to the uprights, which are made of extra heavy proportions; it also largely overcomes the vibration and tendency to rise when running from a wide to a narrow section, so prevalent in the straight type of cross rail. The carriage is 23 inches wide with feed for a cut 8 feet long, is operated by spiral pinion and rack, has variable feed through friction disks, and power quick traverse in either direction by reversing motor. The table itself has flat bearings on the bed, and is of the outside gibbed design.



Newton Slab Milling Machine.

EXPANDING CORE DRILL.

The expanding shell drill shown in the accompanying half-tone is made by the Three Rivers Tool Co., Three Rivers, Mich. This tool is made in various sizes, and is designed to be used on a standard shell reamer arbor for finishing cored



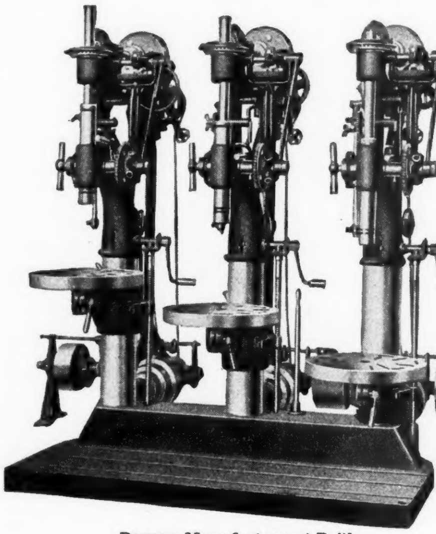
Expanding Core Drill.

holes. The 3-inch size, which is shown in the half-tone, will take a cut ¼ inch deep on a side. As the reamer wears, it may be expanded by screwing in the taper shell, shown at the right-hand end, and then grinding down to size again. When made in solid form for use in the chucking machine, a solid taper plug is used instead of the taper shell for expanding the reamer. The solid reamer or drill may be obtained in smaller sizes than 1 15-16 inches, which is the limit for the shell form.

TWENTY-THREE INCH MANUFACTURERS' GANG DRILL

The B. F. Barnes Co., Rockford, Ill., has added a 23-inch size to the line of manufacturers' drills of which they make a specialty. This size differs from the 14 and 20-inch in that it is regularly built with independent columns and tables, although it will also be furnished with a heavy supporting pillar and single table, after the style of the smaller sizes,

if the customer so desires. The feature of these machines is the fact that for manufacturing work a large number of them may be run by one operator. After the work has been placed in position on the table, the drill is started by the touch of a lever, the spindle advances rapidly until the drill reaches the work, and then the slow feed is thrown in. An automatic

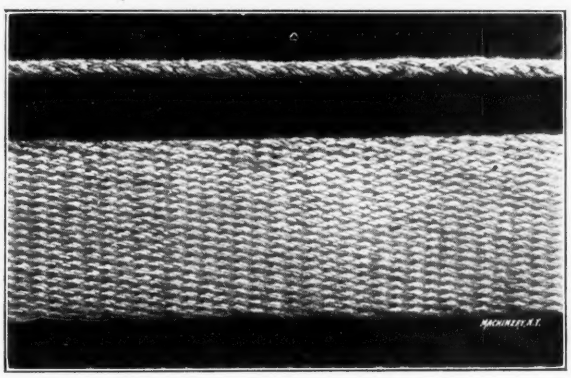


Barnes Manufacturers' Drill.

stop throws out this feed when the cut is completed, and returns the spindle at once to its first position. No attention is required on the part of the operator other than that necessary to keep the machine supplied with work. The 23-inch size, which is here illustrated, is built for considerably heavier work than the other machines of this line. It is strong enough to drive a 1½-inch twist drill in solid material, and will bore out considerably larger holes in cored work. Each spindle has independent feed and independent speed. The individual column style provides for independent adjustable tables for varying heights of work. The tables are regularly furnished in round style, but when oil pump attachment is required, square tables with oil channels will be provided, the only extra charge made being for the oil pump equipment.

A HIGH-SPEED COTTON BELT.

The botton belt of which a section is shown in the half-tone below is designed particularly for use at high speeds, to do away with the trouble experienced in running leather belts over small pulleys revolving at high velocity. It is woven

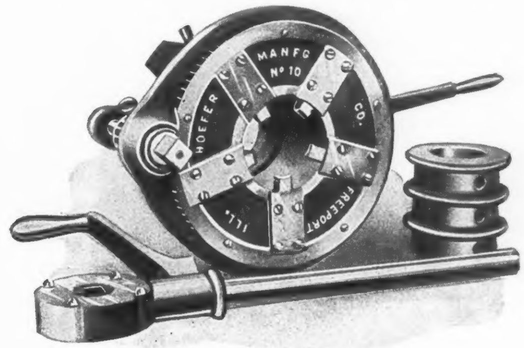


A High-speed Cotton Belt.

endlessly, without lap, or joint, and its makers, the Creamery Belting & Supply Co., Hinsdale, Ill., claim that it will greatly out-last and out-wear any leather belt in the service for which it is designed.

POWER PIPE THREADING MACHINE.

The pipe threading machine shown in the accompanying cut is made by the Hoefer Manufacturing Co., Freeport, Ill. The machine is capable of threading pipe of any material to a standard taper thread, as the dies slightly withdraw as they feed forward, thus increasing the diameter of the thread. The dies feed forward automatically, so the operator does not have to force them forward on to the work, the gear in which the dies move being drawn in toward the work by a threaded sleeve of the same lead as that of the dies. The thread is cleanly cut to the proper taper, and it makes a well-fitted joint, leaving the pipe strong where the thread terminates. A valuable feature of the machine is the provision made for quickly adjusting the thread to a size slightly above or below the standard. Another point is that it requires but one set of chasers (five in a set) for each size of machine to cut the thread



Power Pipe Threading Machine.

on any pipe within its range. Since the dies are not removed from the machine, they are not easily lost. When they become dull they may be readily sharpened by simply grinding the face, and as they are all numbered in their proper order, any number may be duplicated. The revolving parts are so protected that they will not clog with dirt or chips.

WHITNEY CHAIN REPAIR TOOL.

This tool, which is shown in Fig. 1, has been gotten out by the Whitney Manufacturing Co., of Hartford, Conn., to facilitate the repair of their well-known "Whitney" automobile chains. Their detachable roller chain has been adopted by

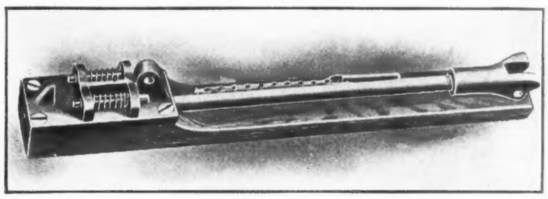


Fig. 1. Whitney Chain Repair Tool.

many of the leading automobile manufacturers. This new device is intended to make the chain more popular from the users' stand point. Every "Whitney" Detachable Chain contains one connecting link, which is distinguished from the

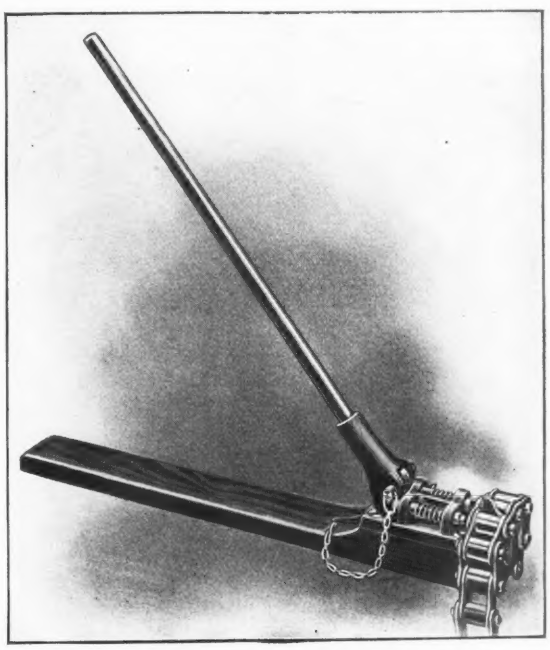
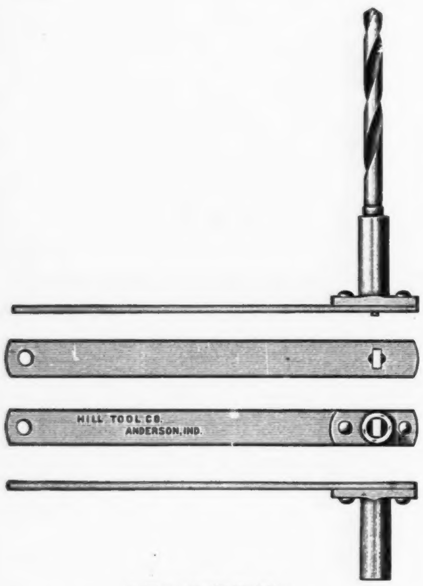


Fig. 2. The Repair Tool at Work.

others by notches in the top surface. This link fits loosely on the rivets and may be removed by hand with ease. All of the other detachable links, however, are purposely forced tightly on to the rivets, and certain users have had some difficulty in removing them on this account. As the rivets are made very hard, the ends will sometimes crack through the cotter pin hole when a user undertakes to detach the chain by means of a hammer or wrench. The new device shown in action in Fig. 2 eliminates all such difficulties, the links being removed with the greatest ease, and with no danger of damaging any of the parts.



A New Drill Holder.

The device will be placed on the market soon after October 1st, and it will be made for all sizes of "Whitney" Detachable Roller Chains.

THE HILL STANDARD DRILL HOLDER.

The Hill Standard Manufacturing Co., of Anderson, Ind., have placed on the market the improved drill holder shown in the accompanying half tone. This holder is made in such a way as to combine lightness and strength. The handle portion of the tool is made of steel, hardened at the end which is riveted to the body of the holder. Through the handle at this point is cut a slot into which the tang of the drill projects, thus preventing it from turning. The fact that the metal around this opening is hardened, lessens the possibility of its being distorted by the torque of the drill. To insert the drill it is simply placed in the holder and given a tap, in which position the tang extends beyond the back face. To remove, the tang of the drill is given a slight blow, thus loosening the drill at once and obviating the use of drift. Should occasion require the use of a drill with the tang twisted off, the countersink on the back side of the holder forms a seat for the lathe center.

TWO SPECIAL GRINDERS.

The Safety Emery Wheel Co., Springfield, Ohio, have recently constructed two special grinders for their customers. In Fig. 1 is shown a surface grinder of the planer type, which was built for the West Milwaukee shops of the Chicago, Milwaukee & St. Paul Railway, to grind their guides and other flat work. The head which carries the wheel of this machine has an automatic reciprocating motion on the cross rails, as well as the usual back and forth movement of the table. This movement may be changed to take place at either the cross rail or the table by the movement of a hand lever. The back and forth movement of the wheelhead is obtained in a simple manner, from a single pulley, which revolves a shaft on which is mounted a set of differential gears which are alternately connected by a clutch to the screw, from which the head takes its motion. The reversing movement is effected automatically, the lateral movement of the wheel head combined with that of the platen tending to keep the base of the wheel true, thus insuring perfect work. All parts of the machine can be operated from the front side. The table has a lip three inches high above the working part of the platen, so that the under side of the work can be buried in water, if desired to keep the work cool. A centrifugal pump furnishes a large supply of water to the wheel to keep the work cool and remove loose

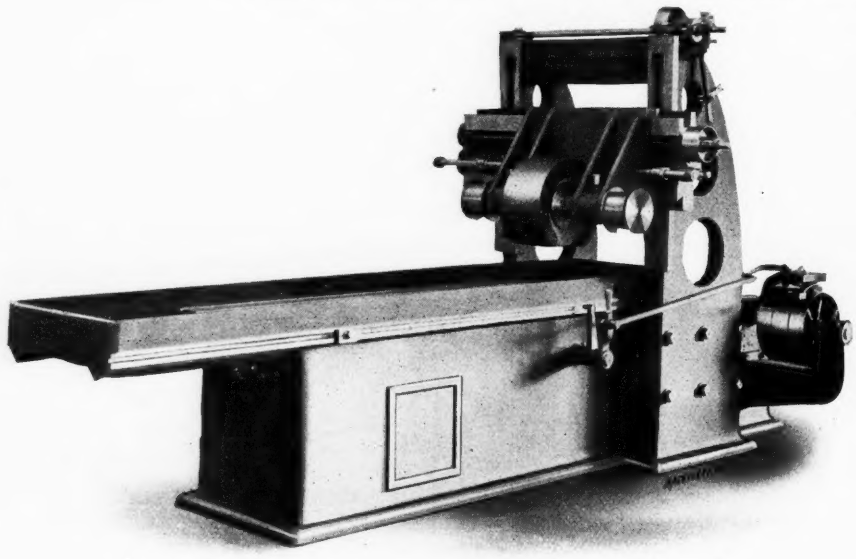


Fig. 1. Planer Type Surface Grinder.

emery. The base of the machine has a solid bottom, thus acting as a reservoir to hold the water and retain the dirt, which may be easily removed therefrom at any time. The machine weighs about 10,000 pounds, the table has a working surface of 18 inches by 72 inches, and will take in work 20 inches high. A longer table will be furnished if desired.

The machine shown in Fig. 2 is a grinder made for the National Tube Co. for finishing the larger sizes of ball dies which are used in making welded tubing. Some of these dies

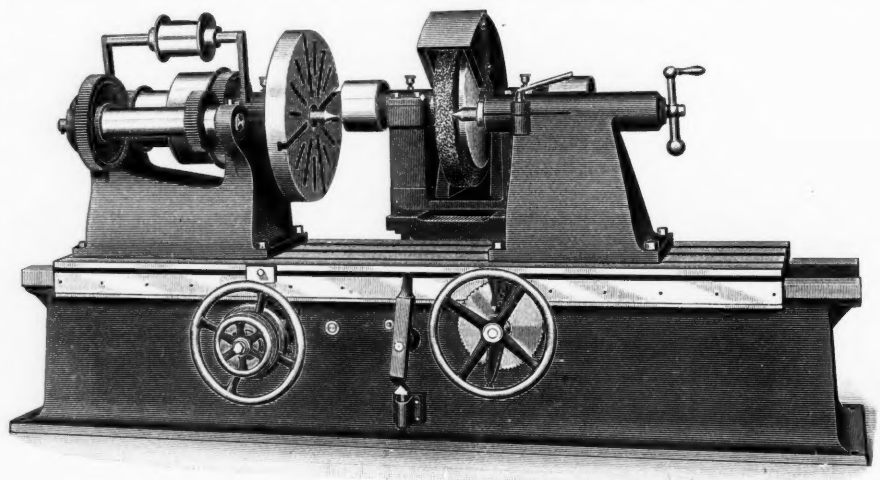


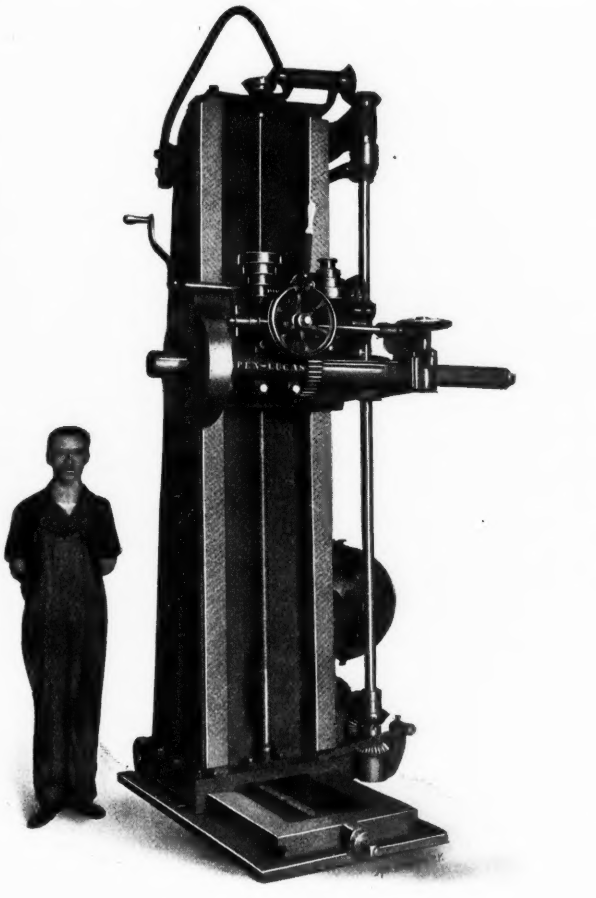
Fig. 2. Special Ball Die Grinder.

weigh as much as 300 pounds each. The machine will take in work 40 inches between centers, 36 inches in diameter. The reciprocating movement for the table is obtained from separate pulleys in the counter shaft. The emery wheel, which feeds automatically toward the work at every stroke of the table, is carried by a carriage at the rear of the machine. The weight of this machine is about 10,000 pounds.

PORTABLE BORING, MILLING AND DRILLING MACHINE.

This machine is intended primarily for use in connection with the floor plate system of building heavy electrical machinery. The example shown in the cut has a vertical travel for saddle of 72 inches, a horizontal movement on the base of

24 inches, and a spindle feed of 24 inches. All these feeds are automatic in either direction, thus making the tool suitable for milling and counterboring as well as for boring and drilling. It is sufficiently heavy and rigid to allow the use of an 18-inch high speed steel inserted tooth cutter head, and the range of speeds and feeds is designed to cover all cases from cutters of this size down to small drills. The spindle is made of hammered crucible steel, 4 inches in diameter, and feeds through

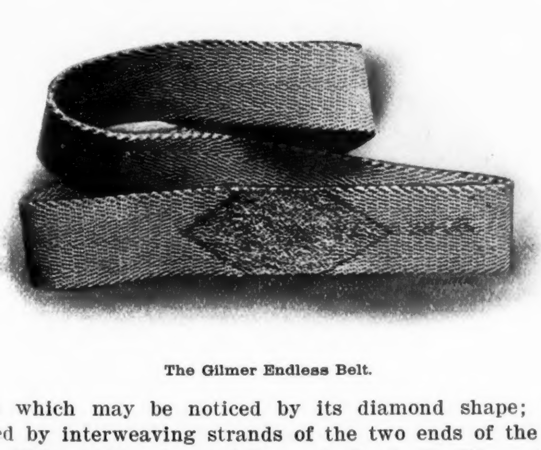


Portable Boring, Drilling and Milling Machine.

a gun metal sleeve. It has a No. 6 Morse taper hole in the end, also a pin hole for retaining the bars and holding tools in place. The machine shown in the cut is driven by a 3 to 1 variable speed Crocker-Wheeler motor. The gearing is made of hammered crucible steel cut from the solid, and bearings are all lined with bronze. The Espen-Lucas Machine Works, Philadelphia, Pa., are the builders.

AN ENDLESS COTTON BELT.

Messrs. L. H. Gilmer & Co., 3952 Market St., Philadelphia, Pa., make a cotton belt for severe service, which is shown in the accompanying cut. The salient feature of this belt is the



The Gilmer Endless Belt.

splice which may be noticed by its diamond shape; it is formed by interweaving strands of the two ends of the webbing. This webbing is specially made for L. H. Gilmer & Co., of high-grade material, and is woven in such a manner that it may be worn nearly half-way through before the separate strands begin to pull apart. It costs less than a leather belt, is more pliable, and is unaffected by oil.

A patent controlled by B. F. Sturtevant Co., of Boston, Mass., has just been issued for a special type of exhaust hood for grinding and polishing wheels. Its special feature consists of a receptacle to catch the particles of solid matter passing from the wheel. The suction being controlled so that it is not quite sufficient to draw them away, these particles fall to the bottom and are there collected, while the practically free air passes through a collector where the last vestige of dust is removed. The receptacle can be readily emptied when it becomes filled, and its use avoids excessive wear on the exhaust fan, piping and collector.

The hood is so designed with hinges and clips that the wheel may be readily removed or adjusted to fit the wheel as it wears to a smaller diameter. The outlet is connected to the exhaust fan, and a shield, a swivel plate and an extension slide may be adjusted so as to more fully enclose the wheel and prevent the discharge or particles into the room.

* * *

CAUTION!

As we go to press our attention is called to the fact that the receipt entitled "To Harden Cast Iron," published under "Machine Shop Receipts and Formulas" in the April number of MACHINERY, is of a dangerous character if improperly compounded. Among the ingredients is sulphuric acid and the poison cyanide of potash. When cyanide of potash and sulphuric acid combine, hydrocyanic acid gas is generated, which is a deadly poison to inhale. Great care should, therefore, be taken not to add the sulphuric acid to the cyanide of potash until the latter has been placed in the ten gallons of water called for by the receipt, so as to dilute the solution and render the generation of the gas less rapid. Under no circumstances should this rule be deviated from.

* * *

THE WARRANT MACHINISTS IN THE NAVY.*

A contributor who signs himself "W. M." gives an explanation of the position occupied by warrant machinists in the navy, the requirements for the position, duties, responsibilities, etc. The advent of the warrant machinist came with the navy reorganization bill of 1899 and there are now 180 such machinists, which, however, is only about half the number actually required to perform the duties devolving upon them. Owing to this scarcity, none have been available for gunboats or destroyers. The following extracts from the navy regulations give an idea of the qualifications:

"Vacancies in the list of warrant machinists shall be filled by competitive examination before a board ordered by the Secretary of the Navy, and open to all machinists in the navy, and to other machinists of good character, not above thirty years of age, authorized by the Secretary of the Navy to appear before the board.

"With applications from machinists in the navy there must be statements of opinion of the commanding officer and engineer officer under whom the applicant is serving. These opinions will be limited to the question of whether the applicant is regarded as qualified for the position of warrant machinist and worthy of such advancement.

"Applicants from civil life must furnish testimonials of good moral character and correct habits, and certificates showing experience in machine shop, and in the engine room of a steamer. . . . No applicant from civil life will be examined who is not a machinist by trade, and has not had the care and management of the steam machinery of a sea-going vessel in regular service."

These examinations, which are held annually in September, embrace a general knowledge of the engineering business, such as is gained by all practical engineers through their trade as a machinist, their earlier experience at sea, and probably through the use of a considerable quantity of "midnight oil" applied as a lubricant to the mental wearing surfaces.

* From Marine Engineering, September, 1905.

Sea Duties.

Warrant machinists act as assistants to the engineer officers of the ship in all that relates to the care and maintenance of the machinery. Machinists are assigned to the different departments of the vessel according to their ranks; senior in rank being in the starboard engine room, second in the port engine room, third in the fire rooms, and the junior has duties in connection with the auxiliary machinery. The warrant machinist must also do watch duty, similar to that performed by the assistant engineer of the mercantile marine. They have to stand regular watches, not more than four hours each. The engine room watch on war ships consists of one warrant machinist, one machinist in charge of each engine, two oilers in each engine room, one storekeeper, and one coal passer, detailed as messenger. In port the warrant machinist performs "day's duty" which begins at 8 A. M. and continues for twenty-four hours, though it is permissible to turn in from 9 P. M. to 5 A. M., but subject to call at any time. If there are four efficient warrant machinists, the practical engineering duties can be well performed without taxing their endurance. The senior engineer officer to whom warrant machinists are responsible is usually a lieutenant-commander or a lieutenant, generally a considerate officer. As a general thing, the warrant machinists are allowed the freedom of their own ideas in pursuing mechanical methods, which their experience indicates to be the most successful.

Shore Duty.

It is an unwritten rule in the navy, in time of peace, that three years' sea duty constitutes a "cruise," after which officers are assigned to shore duty for such period as the exigencies of the service will permit. This period is usually about two years, in the case of most warrant officers, which is an agreeable relaxation from life on board ship. On shore, warrant machinists are assigned as assistants to the engineering officers at navy yards, where their duties consist of detail work in connection with vessels building or under repair. They are also assigned as assistants to the naval inspectors at private shipyards where war vessels are under construction; and they act as assistants to the inspectors of material at various places where naval material is manufactured.

Unfortunately, however, owing to the limited number of warrant machinists in proportion to the demand for their services at sea, they have not enjoyed as much shore duty as other warrant officers. It is hoped steps will soon be taken to remedy this defect through proper legislation. At present the increase is limited by law to twenty appointments yearly. This fact, combined with the knowledge that warrant machinists obtain comparatively little shore duty, tends greatly to retard the healthy growth of the corps.

Promotion.

1. Boatswains, gunners, and warrant machinists having four years' service as warrant officers, and being under thirty-five years of age, are eligible for promotion to ensign, and are then in line of promotion to highest rank.

2. All warrant officers (except warrant machinists, for whom it is hoped the law will provide at an early date) are, after six years' service as warrant officers, eligible for commission as "chief" in their respective grades, "to rank with, but after, ensign."

The examination for ensign embraces navigation, seamanship, ordnance, and steam engineering, and is not beyond the attainment of one having a good common-school education who becomes a warrant officer under thirty years of age. Of ten warrant officers who successfully passed this examination in 1904, three were machinists.

Promotion to "chief" grade is made upon physical and moral fitness; a satisfactory showing in the efficiency reports during the previous six years; and upon passing an examination in a few subjects in which much care, judgment, and responsibility are involved. This promotion carries the distinction of a commission, and the pay of an ensign, but "does not include additional right to quarters, nor to command." Its main advantage is the increase of pay after reaching the maximum, which is shown as follows:

Highest sea pay of chief grade	$1,960
Highest sea pay of warrant officer, with ration	1,908
Increase	$52
Highest shore pay of chief grade, with allowance	$2,248
Highest shore pay of warrant officer, with allowance	1,888
Increase	$360

Pay.

The sea pay of warrant machinists ranges from $1,200 to $1,800 per year, depending at date of appointment upon whether appointed from civil life, or from the navy, with previous service. Those appointed from the navy advance as follows: after three years' service, $1,300; six years, $1,400; nine years' service, $1,600; twelve years' service, $1,800; or maximum pay of warrant officers. Those appointed from civil life are credited by law "at date of appointment, for computing their pay, with five years' service." Thus it will be seen that a warrant machinist appointed from civil life enters on $1,300 per year, and one year later is advanced to $1,400; four years later to $1,600; and seven years later to $1,800. To the yearly sea pay at all times may be added $108 extra allowed for rations. Shore pay is about 10 per cent less than sea pay, but the "allowance" is greater on shore than "rations" at sea.

When "incapacitated for the further performance of duty at sea," officers are retired on three-fourths of their sea pay; and at the age of sixty-two, all officers retire on three-fourths pay.

* * *

The usual method of detecting leaks in the vacuum system of compound engines is to examine the joints by means of a lighted candle. The points where air is drawn into the system will be shown by the flame of the candle drawing inwards. This test, however, is not of the greatest reliability, as it is not delicate enough to show such flaws as a spongy casting which might admit a large volume of air in total, but which is not a localized fault sufficiently great to deflect the candle flame appreciably. One of the "wrinkles" contributed to a paper presented at the June convention of the National Electric Light Association, suggests a method by which such leaks can be detected. It is turning the full steam pressure into the exhaust pipe, between the engine and the condenser. With a steam pressure of, say, 100 pounds, leaks that would defy the candle test are likely to be discovered. With the high vacuum required in steam turbine installations, it is important that the slightest leaks be stopped, and for this reason the scheme outlined by Mr. Hall, Fall River, Mass., is one worth using where leakage is suspected, but cannot be found by the ordinary method.

* * *

It appears from tests that have been made in the Sheffield district that the strength of a grindstone is considerably reduced when wet, as compared with a dry condition. The wetting not only reduces the tensile strength of the stone, but the added weight also, of course, makes the centrifugal stress greater for the same peripheral speed. This reduction of strength as between dry and wet, it appears, amounts to as much as 40 to 50 per cent. For example, a square-inch section stone when dry broke under a stress of 146 pounds, and when soaked in water over night, another piece of the same stone broke at only 80 pounds. In another case, the figures were 186 pounds and 116 pounds, the conditions being the same. Strange to say, there appears to be no settled standard as to safe speeds for grindstones. Some Sheffield grinders run their stones at 4,500 speed per minute, and others limit the speed to 2,500, but the number of breakages do not appear to be greatly influenced by the great difference in speed. It is very probable that a frequent cause of grindstone breakages is the presence of hidden flaws and cracks which increase in extent with use, and as the outside material is removed the strength of the stone is reduced to below the breaking limit.

* * *

Jack had no end of trouble with No. Naughty-naught on his homeward run, says *Railroad Men.* It had been almost impossible to keep up enough steam to make the trip; several parts had been running hot, and in fact everything went wrong. When he arrived at the yard considerably behind